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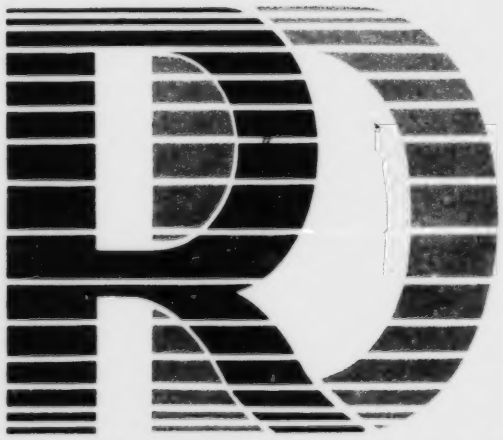

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VIBRATION MONITORING SYSTEM FOR PUMPSETS AT DARLINGTON NUCLEAR GENERATING STATION

Report No 86-68

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Mechanical Research Department

ABSTRACT

This report deals with the design and development of an on-line, computer managed condition monitoring system for major pumpsets at Darlington Nuclear Generating Station using vibration analysis. In total 124 pumps are monitored with 508 sensors. The monitoring system automatically performs baseline comparisons, generates trend plots, and performs vibration signature analysis to provide diagnostic information. The main benefits of the system are realized when it is viewed as a maintenance tool. To maximize these benefits, it is recommended that its use be integrated in the overall maintenance procedures for pumpsets in the plant.

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740627-101-550	825.4	May 8, 1986	86-68-K

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EXECUTIVE SUMMARY

VIBRATION MONITORING SYSTEM FOR PUMPSETS AT DARLINGTON NUCLEAR GENERATING STATION

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Mechanical Research Department

A computer managed vibration monitoring system for all major pumpsets at Darlington NGS has been developed. In total 124 pumpsets are monitored with 508 sensors. The system is designed to detect deterioration in machinery condition at an early stage, to identify probable cause(s), and to complement the function of an existing primary protection and alarm system.

The monitoring system described in this report automatically performs base line comparisons, generates trend plots, and performs vibration signature analysis to provide diagnostic information.

The system developed in this work provides a powerful tool for on-line condition monitoring and diagnosis of pumpsets in a CANDU power plant. It has the potential of improving plant availability and reducing maintenance cost. The system can easily be adopted to monitor other rotating machinery by creating a suitable data base.

The main benefits of the system are realized when it is viewed as a maintenance tool. To maximize these benefits, it is recommended that its use be integrated in the overall maintenance procedures for pumpsets in the plant.

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TABLE OF CONTENTS

Page

1.0 INTRODUCTION

2.0 DEFECT MECHANISMS

3.0 SENSOR SELECTION AND LOCATION

4.0 SYSTEM OPERATION

- 4.1 Data Acquisition
- 4.2 Data Manipulation and Reduction
- 4.3 Comparator
- 4.4 Diagnostic System
- 4.5 Output Module

5.0 HARDWARE IMPLEMENTATION

6.0 SOFTWARE DESIGN CRITERIA

7.0 SOFTWARE CONFIGURATION

- 7.1 Software Capabilities
- 7.2 Implementation of On-Line Diagnostics

8.0 SYSTEM TESTING

9.0 CONCLUSIONS

10.0 RECOMMENDATIONS

11.0 ACKNOWLEDGEMENTS

REFERENCES

APPENDIX A INSTRUMENTATION LAY OUT

APPENDIX B IMPLEMENTATION OF DIGITAL FILTERS

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To Mr. M.A. Machacek
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Darlington Engineering

VIBRATION MONITORING SYSTEM FOR PUMPSETS AT DARLINGTON NUCLEAR GENERATING STATION

1.0 INTRODUCTION

Condition monitoring of power plant machinery using vibration, flow, pressure, temperature and other process variables is a well established practice/1,2,3/.

Until recent years, the general technique for continuous vibration monitoring in power plants consisted primarily of a transducer detecting either bearing housing vibration level or relative displacement between shaft and bearing, overall amplitude monitor, and an alarm triggering system. The main purpose of such a "primary" system is protection and alarm.

Although there are exceptions, experience has shown that the overall vibration level is not an effective criterion for early detection of deterioration in machinery condition. The data is highly scattered, and it is difficult to discern consistent trends with time. Also, overall vibration levels provide no diagnostic information. Spectral analysis performed in an off-line mode is very laborious.

The system described in this report is designed to perform on-line monitoring and diagnostics of major pumpsets on all units at the Darlington Nuclear Generating Station. This "secondary" system is also designed to complement the function of a "primary" protection and alarm system by providing early detection of impending defects. This report provides an overall engineering description of the vibration monitoring system and its capabilities. Two secondary reports, B86-17-K the User's Manual and B86-15-K, the Software Manual, provide detailed instructions on how to operate the system and how the computer software has been assembled.

There are two main approaches taken in the development of such an on-line diagnostic system. The first approach employs pattern recognition techniques/4,5/. In this approach, features are extracted from the data signal vectors and then are interpreted (classified) using pattern recognition techniques to

detect wave form abnormalities reflecting machine malfunctions. There are many different classification techniques available; each one is based on a different set of mathematical assumptions. Examples of these algorithms include the empirical Bayes procedure, K-nearest neighbour algorithm and least square procedures/6/. While the pattern recognition approach is attractive because it requires little knowledge of the machine under investigation, its implementation requires the accumulation of a large amount of data for normal and abnormal conditions which may not always be practically feasible.

The second approach to on-line diagnostics involves the creation of a knowledge base relying on the accumulated experience and knowledge of vibration analysts. The most straightforward realization of such an "expert" system is the creation of a table of known symptoms (frequencies and/or time domain descriptors such as the kurtosis value, crest factor, etc) generated by the various defects under consideration. The diagnostic function is then achieved by comparing say specific frequency/frequencies of a given machine under surveillance against the defect identification table.

1.1 Purpose of the System

i) To detect pumpset condition deterioration or defects at any early stage by base line comparison of vibration spectra.

ii) To determine the cause of high vibration levels or changes in vibration spectra by comprehensive time and frequency domain vibration signature analysis.

iii) To provide first and second level alarms and diagnostic information when limits are exceeded.

iv) To store pumpset vibration data and provide trend plots.

1.2 Benefits

i) Compared with monitoring overall vibration levels and occasional manual spectral analysis, the proposed system detects a larger percentage (estimated at 3 times) of machine deterioration (defects) and at an earlier stage.

ii) Identifies which machinery requires what maintenance, reducing unnecessary maintenance and costs.

iii) Provides feedback on effectiveness of maintenance.

iv) Minimizes vibration monitoring labour costs.

v) Improves plant equipment availability, reliability and safety.

2.0 DEFECT MECHANISMS

Failure modes on similar pumpsets at Pickering NGS 'A' were examined using various time and frequency domain analysis techniques. A number of tape recorded case histories of pumpset failures were analyzed and the results were reported in detail elsewhere/7/.

The objectives of this analysis were mainly to investigate how these failure modes manifest themselves in the time and/or frequency domains and to evaluate the effectiveness of various signature analysis methods in detecting them.

Several typical defects involving different types of pumpsets will be discussed in this section.

Case 1

The application of the synchronized time averaging technique to detect a cracked gear in a condenser cooling water pump is illustrated in Figure 1. Since this pumpset has two shafts, it has one velocity pick-up mounted at the top of the motor and one near the pump shaft seal below the gearbox. A phase marker is also installed on both the motor shaft and the pump shaft. The damped wave trains (one revolution of pump shaft apart) in Figure 1 arise from shock excitation of a structural resonance at 160 Hz. The shock excitation is caused by the discontinuity of a cracked tooth passing through the gear mesh. The progression of deterioration in gear condition is seen from the consistent increase in the magnitude of the highest peak in the wave train, as illustrated in records "a" to "d". Compared to record "a", this magnitude increased by 41% in record "b", by 83%, in record "c", and by 91% in record "d" at the time of failure.

The corresponding increase in the overall level of the vibration signal between records "a" and "d" was 31%. Thus, the synchronized time average peak is approximately three times more sensitive in detecting the cracked gear tooth than the overall vibration level in this case. Also, the synchronized time average analysis provides information about the type of defect whereas the overall level provides no diagnostic information.

Case 2

The detection of a pinion tooth fracture in a condenser cooling water pump using frequency analysis is demonstrated in this case.

Two records obtained from a velocity transducer mounted on the pump gland (Unit 3, Pump 1), were analyzed. In this case the tooth meshing frequency is 860 Hz. The sidebands seen in Figure 2 indicate a modulating frequency of 20 Hz (motor shaft). The value of the modulating frequency is confirmed by the cepstrum analysis in Figure 3.

The deterioration in the pinion condition is manifested in the frequency spectra mainly as a change in the strength and the number of sidebands around the tooth meshing frequency and its harmonics.

Case 3

This case illustrates the use of frequency analysis to identify a defective bearing in a moderator circulating pump. The analysis was performed on the signal obtained from a velocity pickup mounted on top of the pump motor. The frequency spectra are shown in Figure 4. Records "a" to "c" show a consistent increasing trend in the magnitude of the second harmonic of the inner race ball pass frequency of the lower motor guide bearing (deep groove ball bearing). This magnitude increased from a value of 0.013 mm/s pk in record "a", to 0.016 mm/s pk in record "b" (14.3% increase), and reached 0.025 mm/s pk in record "c" (78.5% increase). The spectra also show a substantial growth in a number of high frequency components which is usually an indication of bearing deterioration. Also, since these frequencies occur at other than multiples of the motor operating speed, the order domain analysis indicated a growth in the non-harmonic power from 0.977 $(mm/s)^2$ in record "a" to 59.003 $(mm/s)^2$ in record "c", a 5939% increase. The overall level of the vibration signal increased from 2.76 mm/s pk in record "a", to 5.15 mm/s pk in record "b", a 86.59% increase, to 18.8 mm/s pk in record "c" representing a 581.16% increase when compared to record "a". Thus, the non-harmonic power is a more sensitive indicator of the bearing deterioration than the overall level which would allow detecting the defect at an earlier stage. Also, the overall level results provide no diagnostic information while the frequency analysis indicates the likely cause of increased vibration.

In addition, the spectral autocorrelation analysis results presented in Figure 5 clearly show significant growth in the

outer and inner race ball pass frequencies as well as the fundamental train frequency of the defective bearing. This provides further assistance in defining the nature of the defect.

Case 4

Figure 6 shows a spectral auto-correlation analysis performed on the velocity signal obtained from a transducer mounted on top of the motor of a vertical low pressure service water pump. In this case, a fatigue fracture of the pump drive shaft manifested itself in the disappearance of side bands around the roller pass frequency and its harmonics of the outer race of the top motor thrust bearing. The total power in the signal increased from 9.638 mm^2 in record (a), to 36.401 mm^2 in record (b), to 45.496 mm^2 in record (c), and finally reached 954.550 mm^2 at the time of failure. This case also illustrates that the disappearing as well as the growth of roller pass frequency side bands can be indicative of machine condition deterioration.

Case 5

The detection of an impeller vane fatigue fracture in a vertical low pressure service water pump is shown in Figure 7. In this case, three records were taken from a velocity pickup mounted on top of the pump motor. Records "a" to "c" show a consistent increasing trend in the power of the fundamental rotational frequency from 0.58 $(mm/s)^2$ in record "a" to approximately 58.5 $(mm/s)^2$ in record "c".

Case 6

The use of the high frequency resonance technique in detecting bearing defects is illustrated in this case. The results shown were obtained from a bearing test rig (see Figure 8) which was used to evaluate the performance of various vibration signature analysis techniques for defect detection in rolling element bearings/8/. Various defects were artificially induced in the tested bearings, and the resulting vibrations monitored using a piezoelectric accelerometer mounted on the bearing housing.

In the case presented here, a crack clear through the inner race was induced. The band-pass filter in the HFRT algorithm was centred at 19.5 KHz (one of the bearing system natural frequencies). The bandwidth used was $\pm$ 2.5% of the centre frequency. The resulting vibration spectrum is shown in Figure 9, while the output of the HFRT algorithm is presented in Figure 10. Referring to the later figure, the defective bearing

frequencies are clearly identified (ball pass frequency of the inner race, BPFI, and its harmonics.

Further discussion of the HFRT algorithm can be found in Section 4.4.

3.0 SENSOR SELECTION AND LOCATION

The type of sensor which will give best results depends on what condition it is desired to monitor and protect against or what malfunctions one wishes to analyze. Due to cost restraints and lack of access to various machine locations, it is normally necessary to compromise and select the minimum number of transducers with which the objectives may be achieved. The most desirable sensor locations are areas of maximum response and near sources of possible excitation or malfunction. These locations are determined from theoretical rotor dynamics analysis results, experience with similar machinery and vibration measurements taken during commissioning.

In general, velocity transducers and/or accelerometers were mounted close to antifriction bearings and to gearboxes (if any). Two displacement probes installed in the same radial plane, and separated by 90 degrees, were typically mounted at pump seals, motor and pump inboard and outboard bearings. In addition a phase marker producing a pulse per pumpset shaft revolution was also mounted close to the exposed part of the shaft, coupling, etc.

The selection and location of the vibration transducers were determined by Darlington Engineering Department/9/. As an example, the instrumentation layout of the main moderator pump is shown in Figure 11.

Appendix A provides the detailed instrumentation lay-out for all pumpsets included in the vibration monitoring system.

In order to provide a comparison between the vibration monitoring systems at both Pickering NGS 'A' and Darlington NGS, the instrumentation lay-out, and the vibration monitoring system at Pickering NGS 'A' are shown schematically in Figures 12 and 13 respectively.

4.0 SYSTEM OPERATION

The main activities of the vibration monitoring system are illustrated in Figure 14. A detailed flow diagram of the system operation is given in Figure 15. As seen in these two figures,

the architecture of the system is divided essentially into five main functions:

1. Data acquisition
2. Data manipulation and reduction
3. Comparator
4. Diagnostic
5. Output module

4.1 Data Acquisition

The outputs of the vibration sensors are first multiplexed. The scanning cycle (time required to complete acquisition and analysis of all sensor signals) is set at 12 hours and is controlled by the mini-computer. The phase marker signal is used to ascertain whether the pump is running. The output signal of the multiplexer is then amplified and low-pass filtered before being digitized for data reduction and analysis. The operation of the multiplexer, amplifier, and filter are controlled by the mini-computer. In order to validate the signal, its harmonic power is calculated and compared against a predetermined lower bound (based on experience).

4.2 Data Manipulation and Reduction

A set of features is calculated for each signal as well as an averaged FFT spectrum (eight averages for acceleration or velocity signals, and four averages for displacement signals). The signal features are:

- Overall level, which is a measure of the vibration signal intensity.

- Total harmonic power, which provides a measure of the harmonic component in the vibration signal (ie, related to rotational speed and its harmonics).

- Total non-harmonic power, which provides a measure of the non-harmonic component (ie, related to antifriction bearings, rubs, etc).

- Magnitude at first order, which is an indication of rotating elements out of balance; the most common cause of excessive vibration in machinery.

- Phase at first order.

- Overall magnitude of all orders from 0.0 to 0.85.

- Overall magnitude of all orders from 1.2 to 1.8.

- Overall magnitude of all orders from 1.8 to 2.2.

- Overall magnitude of all orders from 2.2 to 2.8.

- Overall magnitude of all orders from 2.8 to 3.2.

- Magnitude at number of Vanes x RPM (if any), which is related to the Vanes (blades) passing frequency.

- Total power in the 500-2000 Hz range, which is usually an indication of anti-friction bearing condition.

These signal features are stored for trending on a short (per scanning cycle), medium (per week), and long (per month) term basis for all vibration signals. The averaged FFT spectrum is stored temporarily for subsequent analysis.

4.3 Comparator

Each spectrum is compared to a previously stored base line spectrum for changes in amplitude of discrete frequencies above a predetermined threshold. Due to even relatively small machine speed changes between measurements, a direct comparison of the spectra is usually not satisfactory and it is necessary to normalize their frequency to a machine speed order. If the comparison results in any frequency component(s) exceeding the base line by more than a predetermined threshold value, a consistency check is performed (ie, a new signal is acquired and the previous steps repeated). The concept of base line comparison is illustrated in Figure 16.

4.3.1 Generation of Base Line Spectra

Base line vibration data are those data or sets of data measured or observed when the equipment is known to be operating acceptably. All subsequent measurements will be compared to these base line values to detect vibration changes. Base line data must accurately define the initial stable vibration condition of the machine operating in its normal operating mode and flow rate.

For new and overhauled equipment, there is often a wear in period as illustrated in Figure 17. It is not uncommon to see a change in vibration level during the first few days or weeks of operation/10/. Data taken for base line must define and reflect the zone in which the machine vibration exists.

Base line data should be collected during commissioning and updated after each maintenance action. In addition, as explained above, base line data should be updated depending on the zone in which the machine vibration exists.

4.4 Diagnostic System

The monitoring system performs diagnostics of the vibration signature automatically. The messages indicating the fault type are stored in a knowledge based frequency identification table. This table is created to reflect the appropriate defect identity associated with various frequencies present at levels exceeding preset warning and/or alarm levels. To accommodate small shifts in machine speed, a frequency F in the identification table is defined to exist from $F - \Delta f$ to $F + \Delta f$ where Δf is the frequency resolution.

Once a frequency is identified in the frequency identification table, an attempt is made to match it with an appropriate diagnostic message. Diagnostic messages can be associated with a number of discrete frequencies from any one or more vibration transducers in a given pump.

4.4.1 Generation of Frequency Identification Tables

Table 1 provides an example of a typical frequency identification table for a given pumpset. There are a number of charts of this type available in the references listed in this report. These charts are no substitute for experience and engineering judgement.

On a per pumpset basis, these tables should be generated during commissioning and reviewed periodically using the knowledge and experience gained from the monitoring system. The size of a typical table will also increase with time due to the experience gained with the particular pumpset under surveillance.

4.4.2 Signature Analysis Techniques

In addition, the user has the option of using various diagnostic techniques (user interrupt mode) to identify the defect(s). These techniques are described in the following:

4.4.2.1 Synchronized Time Averaging

The technique is used for extracting periodic components from a complex signal. It is a coherent technique, necessitating a priori knowledge of the extracted component's period, T. It is based on averaging of points one period apart. This enhances

the periodic components relative to the random components (noise).

Let M be the number of samples per period, then:

$$\overline{X}_i = \frac{1}{n_{rev}} \sum_{N=1}^{n_{rev}} X\,(i\Delta t + NT) \qquad (1)$$

for i = 0,1,2,...., M.1

where;

$X(i\Delta t)$ are the time domain sampled values

$\overline{X}_i$ are the averaged values
Δt is the sampling resolution
n_{rev} is the number of revolutions

and,

$$T = M.\Delta t = \frac{1}{f_0} \qquad (2)$$

where f_0 is the frequency of rotation.

Regardless of the magnitude of the repetitive component, its presence can be enhanced by time-averaging the wave form.

This time averaging technique is equivalent to applying a comb filter to the original signal which passes only the fundamental frequency and its harmonics.

The fundamental period, T, is usually based on a synchronizing signal triggered by the main rotating shaft (phase marker).

4.4.2.2 Order-Domain Analysis

The use of order-domain rather than the more familiar frequency domain for the analysis of vibration signals from rotating machinery simplifies interpretation of the results. The basic relationship used for the conversion between the two domains is:

$$O = \frac{f}{f_0} \qquad (3)$$

where f_0 is the fundamental rotational frequency.

Integral orders (Q = 1,2,3, etc) occur at harmonics of the running speed. If the vibrational signal is analyzed for N_{rev} revolutions, then the minimum order resolution achievable is:

$$\Delta Q = \frac{1}{N_{rev}} \tag{4}$$

4.4.2.3 Spectral Autocorrelation

In rotating machinery, the periodic phenomena with harmonic frequencies related to the speed of the rotor and non linearities in energy generation and propagation will introduce energy at frequencies which are functions of the driving frequency. Several techniques from communication theory are extremely useful for identifying harmonic content when the distribution of energy possesses periodic character.

Spectral autocorrelation, wherein the logarithmic amplitude spectrum is treated like a time series, provides information about periodic character in the same manner as its time domain counterpart.

The procedure used to calculate the spectral autocorrelogram is:

(a) The frequency spectrum X is centered by removal of a "moving" mean. This is accomplished by using an auto-regressive low-pass filter of the following form to smooth the spectrum

$$Y(N) = a \cdot Y(N-1) + (1-a) \cdot X(N) \tag{5}$$

where X is the original spectrum (magnitude in decibels), Y is the smoothed spectrum, and a is the filter coefficient. A value of a = 0.9 is used.

(b) The smoothed spectrum is subtracted from the original spectrum and the result X*(N) is autocorrelated.

$$X^*(N) = X(N) - Y(N) \tag{6}$$

(c) The routine used in the computation of the autocorrelogram is:

$$R(LAG) = \frac{1}{f_{max}-LAG} \sum_{N=1}^{f_{max}-LAG} X^*(N) \cdot X^*(N + LAG) \tag{7}$$

where X*(N) is the magnitude, in decibels, of the Nth component of the spectrum.

The series of autocorrelation estimates R(LAG) is normalized by division by R(0), the mean square value. Usually a small number of lags is sufficient. Each lag corresponds to the frequency resolution of the original spectrum.

4.4.2.4 Cepstrum Analysis

The cepstrum $C(\tau)$, defined by Bogert et al/11/, as the power spectrum of the logarithm of the power spectrum, is also a measure of periodicity. Thus,

$$C(\tau) = \left| F_{xx} \{\log F_{xx}(f)\} \right| \qquad (8)$$

where

$$F_{xx}(f) = \left| F\{f_x(t)\} \right|$$

The independent variable τ of the cepstrum is known as "Quefrency", and has the dimensions of time. A crucial factor for the successful application of the cepstrum is sufficient resolution in the original spectrum.

4.4.2.5 High Frequency Resonance Technique (HFRT)

The HFRT process involves the envelope detection, or demodulation, of a high frequency vibration signal which has been filtered by a narrow band pass filter at a specific frequency of interest. This frequency of interest is generally some structural resonance which has been found analytically or experimentally to be the optimum frequency for envelope detection. In some cases a band-pass filter tuned to the natural frequency of the transducer (accelerometer) is a good choice. It will preserve the ringing response of the transducer and eliminate noise data.

The bandwidth of the band pass filter must be at least double the defect frequency of interest ($\pm$5 per cent is usually sufficient) in order to pass the necessary side bands of the carrier frequency. If the bandwidth is set too large, unnecessary noise and distortion is liable to be passed by the filter, along with the carrier and its sidebands.

The envelope detected signal is filtered for the presence of some frequency whose amplitude is indicative of the operating condition of the dynamic machinery element being examined. As

an example, in the case of rolling element bearings, the amplitude of the vibration component at ball-pass or roller-pass frequencies may be examined in this manner.

The HFRT algorithm flowchart is given in Figure 18. The HFRT program has been included in the system software.

The required filters are of the butterworth type (see Appendix B). The filters are first designed and then applied to as many continuous records as required.

For subsequent processing, the lowpass output can be subsampled at a ratio:

$$F = \frac{F_L}{F_S} \tag{9}$$

where F_S is the sampling frequency and F_L is the lowpass cut off frequency.

4.4.2.6 Kurtosis Value

The Kurtosis Value "K" is the fourth central moment of a Gaussian distribution and is a measure of its peakedness:

$$K = \frac{\frac{1}{N} \sum_{i=1}^{N} (X_i - \mu)^4}{\tau^4} \tag{10}$$

where

X_i are the time domain sampled values,
N is the number of data points,
μ is the mean value,
τ is the standard deviation on RMS for a zero mean signal.

The Kurtosis Values in selected frequency bands are used as a measure of bearing conditions. The discrete impulsiveness normally associated with damage emerges in different frequency regions with varying degrees of damage. A Kurtosis Value greater than three is an indication of impending failure.

It should be noted that since the main assumption here is that the vibration signal is represented by a Gaussian distribution, data below 2 kHz should be disregarded since it is heavily

contaminated by the harmonic component(s) arising from the rotating machine.

4.4.3 Establishing Vibration Limits

The American National Standard guidelines for vibration monitoring of rotating equipment in nuclear power plants, ANSI/ASME OM-4-1984/10/, recommends the following procedures for establishing vibration limits. They are consistent with the ASME code, Section XI, IWP, which allows three ranges of vibration; "acceptable range", "alert range" and "action-required (alarm) range". Maintenance of the equipment should occur prior to reaching the lower limit of the "action-required (alarm) range".

The recommended procedure for establishing limits for each of these ranges is:

i) First, use the manufacturer's data if available.

ii) If the manufacturer's data is not available and Section XI, IWP-3000, of the ASME Code applies, Table IWP-3100-2, allowable ranges of test quantities of IWP-3000, should be used to establish the limits for each of the ranges.

iii) When limits for each of the zones cannot be established using either of the above methods, a factor of two increases in vibration should be used as an upper limit for the normal range baseline comparison and a factor of four increase in the baseline vibration for the upper limit for the "Alert Range".

Obviously, the vibration limits will determine the system sensitivity to defect detection. In setting these limits, a compromise between sensitivity to defect detection and minimization of false alarms is required. To increase the flexibility of the user in setting vibration limits, up to ten different limits can be set in the frequency range of analysis.

It should be expected that these vibration limits would be modified with experience in using the monitoring system.

Under normal operating conditions, the system continuously displays on the monitor the last twenty transducers scanned and their status.

If a warning and/or alarm condition is detected, the output module is activated to inform the operator about the condition, based on the results of the diagnostics. A hard copy is

produced automatically defining the diagnostic message. A typical message is shown in Figure 19. In addition, short term trend plots (last 60 scanning cycles) of the signal features are produced. Also, in addition to these plots, which are generated automatically, the user has the option (in the user interrupt mode) to generate medium and long term trends, base line versus current FFT spectra, and plots of the results obtained from using any or all of the diagnostic techniques described earlier. There plots are available both as hard copy and as a screen representation.

5.0 HARDWARE IMPLEMENTATION

Figure 20 identifies the main components of the hardware, and illustrates the data and control flow through the primary and secondary system. Only the hardware implementation for the secondary system is discussed here.

The hardware for the secondary system consists of a DEC PDP 11/23 computer with one mega byte of memory, a 30 mega bytes Winchester disk with a tape cartridge subsystem, a digital FFT analyzer, multi-point scanners and input-output peripherals. The major functions of the computer are to communicate and control the devices which perform the data acquisition and to handle their output. A total of nine multi-point scanners, each with 64 input channels, are controlled by the computer with the use of two parallel interfaces. A maximum of 128 phase and 448 vibration signals are multiplexed into two output channels (one phase and one vibration) connected to the digital FFT analyzer. More input channels can be achieved by stacking up additional multi-point scanners into the system.

The single channel digital FFT analyzer is a sophisticated instrument incorporating a microprocessor, an amplifier, a low pass filter, a 12-bit analog to digital converter and 96 K bytes of internal memory. The low pass filter is a selectable high precision elliptical filter used to eliminate the aliasing error. The analog to digital converter has 12 bit accuracy. The analyzer is capable of digitizing data at a maximum rate of 256 k points per second and up to 16 k points of continuous time data can be sampled and stored in the memory. The Fast Fourier Transform is done in parallel with the digitization process. The use of the FFT analyzer as the data acquisition unit not only provides us with a very fast FFT calculation, it also creates a multi-processing environment for the system. While the computer is examining the FFT data or performing diagnostics, the FFT analyzer can proceed and carry out the digitization and FFT computations for the next vibration signal. The phase signal is routed to the input of the real

time clock in the computer to calculate the machine running speed.

The alarm system has four analog output lines from the digital to analog converter in the computer. With the combination of different voltage levels on each line, the alarm system can identify the pump with excessive vibration to the operator in the control room which is some distance away from where the monitoring system is located.

6.0 SOFTWARE DESIGN CRITERIA

In designing the software, three main criteria were identified. The first criteria was to allow two separate modes of operation: continuous monitoring and user interactive. The continuous monitoring mode is totally automatic and runs without any user intervention. This mode scans all sensors, performs all base line comparisons, trending, and diagnostics. The interactive mode allows interruption of the continuous monitoring and enables the user to review a particular pump status, examine the signal from a specified sensor in the time and/or frequency domains, and review trends.

As a second design criteria, the user interactive mode of operation had to be user friendly. The use of the software should not require a computer specialist and a minimum of training should be required to operate the system.

The final criteria was system flexibility and efficiency. The software makes efficient use of the minicomputer and allows flexibility in the creation and modification of system parameters. This flexibility should also allow the monitoring system to be applied to other environments by creating a suitable data base (ie, other CANDU power plants).

7.0 SOFTWARE CONFIGURATION

In order to address the desired design criteria, the software was implemented in a modular file, linked and controlled structure, as illustrated in Figure 21. The two modes of operation defined in the previous section were implemented as four separate main programs.

The first is the "commissioning" program which allows the user to create the knowledge base. This program also allows the system the flexibility of operating in scaled down or scaled up versions for different operating environments.

The continuous monitoring mode is implemented through one independent "Monitoring and analysis" program controlled by the files created during commissioning. The user interactive mode is implemented using two separate programs mainly to alleviate limitations in maximum allowable program memory. One program of the user interactive mode performs all functions which utilize memory intensive routines such as the digital filter and the digital Fast Fourier Transform. The second program will generate long and short term trends, and perform various user requested analyses on specified transducers. The software is menu-driven in the user interactive mode, facilitating its use by non computer specialists.

The software system utilizes over 20 direct access files which are function specific. Since the major consideration in the design of the files was speed and efficiency of use, direct access files were used.

7.1 Software Capabilities

The software has been designed to handle up to 10 pumps for 21 different pump types, from as many as 8 operating units with up to 9 vibration transducers per pump. The software also supports computer control of the multiplexer, the FFT analyzer, and the eight pen plotter.

7.2 Implementation of On-Line Diagnostics

The heart of the diagnostic function is the frequency identification tables which represent the knowledge base of the system. The diagnostic information associated with particular frequencies are examined on both a per transducer and a per pump basis. The knowledge base is both expandable and modifiable at any point during operation. The learning mode is external. However, the program structure allows new information to be incorporated directly into the knowledge base.

The diagnostic system allows up to 30 frequencies to be identified per transducer. Additionally one can identify a number of frequencies from different transducers on a given pump and associate these with a diagnostic message. Up to 15 records each containing 20 frequencies from various transducers can be so identified on a per pump basis. These capabilities are realized by two direct access files consisting of a linked list for single frequency diagnostics (per transducer) and a linked list for multiple frequency and transducer diagnostics (per pump), as illustrated in Figure 22. This procedure allows the program to model the decision and analysis procedure followed by a vibration analyst. Linked list files were employed to improve

data access speed and to allow simple addition and deletion of records without data rearrangement.

Data is sorted in the order of insertion into the lists. The retrieval of data requires only a straight forward calculation of the starting address and a sequential search along the list with the use of record pointers.

8.0 SYSTEM TESTING

The performance of the monitoring system is currently being tested. The testing program has two objectives. The first is to test the overall performance of the hardware and software. The second objective is to assess the fault detection and diagnostic capabilities of the knowledge base.

The monitoring system is to function without operator intervention and thus it is a necessity that the system be reliable. All software modules have been tested individually and as part of the system. The hardware functions such as multiplexing and data acquisition have also been tested extensively. Further tests will be performed upon system installation in the plant.

Preliminary fault detection and diagnosis tests were performed using a laboratory rotor test rig. Early results indicate that the system is successful in identifying bearing faults and detecting deterioration at an early stage. Further tests will be performed utilizing a full scale pump prototype. Final tests will occur during commissioning and installation of the system in the power plant. This will ascertain the effectiveness of the knowledge base as tailored for the generating station's environment.

9.0 CONCLUSIONS

9.1 A computer-managed vibration monitoring system for major pumpsets at Darlington NGS has been developed. The monitoring system automatically performs base line comparisons, generates trend plots and performs time and frequency domain analyses on the vibration signal to provide diagnostic information.

9.2 The system developed in this work provides a powerful tool for on-line condition monitoring and diagnosis of pumpsets in a CANDU power plant. It has the potential of improving plant availability and reducing maintenance costs. The system can easily be adapted to monitor other rotating machinery by creating the suitable knowledge base.

9.3 In setting the vibration limits during base line comparison a compromise between sensitivity to defect detection and minimization of false alarms is required. Guide lines for setting these limits are included in the report. These vibration limits should be modified with the experience gained from using the monitoring system.

9.4 Base line data should be updated after each maintenance action, and depending on the zone in which the machine vibration exists.

9.5 The frequency identification tables should be generated during pumpset commissioning and reviewed periodically using the knowledge and experience gained from using the monitoring system.

9.6 The efficiency of the vibration monitoring system should improve with time. Correct adjustment of the vibration limits and the generation of the "optimum" frequency identification table will require some experience with using the system at the power plant.

9.7 The main benefits of the system are realized when it is viewed as a maintenance tool. The system is not intended to provide protection against malfunctions, rather the purpose is to assist station personnel in predictive maintenance by identifying impending defects at an early stage and providing diagnostic information.

10.0 RECOMMENDATIONS

To maximize the benefits of this system, it is recommended that its use be integrated in the overall maintenance procedures for pumpsets in the plant.

11.0 ACKNOWLEDGEMENTS

The work described in this report was carried out on behalf of Ontario Hydro's Darlington Engineering Department. The help of Mr. T. Karaim is gratefully acknowledged. The authors would also like to acknowledge the assistance and expert advice of Mr. J. Rosenberg of the Instrumentation and Control Department.

Mr. H.J. Tait made very significant contributions to the software development while with Ontario Hydro.

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6. Doctor, P.G., Harrington, T.P., and P.H. Hutton, "Pattern Recognition Methods of Acoustic Emission Analysis", Pacific Northwest Laboratory, NUREG/CR-0910, PNL-3052, July, 1979.

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10. American National Standard Guidelines for Vibration Monitoring of Rotating Equipment in Nuclear Power Plants ANSI/ASME OM14-1984.

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TABLE 1

VIBRATION TROUBLE SHOOTING CHART

	FREQUENCY OF VIBRATION (Hz = rpm/60)	DIRECTION	REMARKS
ROTATING MEMBERS OUT OF BALANCE	1X rpm	Radial	The most common cause of excess vibration in machinery.
MISALIGNMENT AND BENT SHAFT	Usually 1X rpm Often 2X rpm Sometimes 3X & 4X rpm	Radial and Axial	The second most common fault.
RUBS	1X rpm and possible higher multiples	Radial	A common cause of excess machinery vibration.
DAMAGED ROLLING ELEMENT BEARINGS (Ball, Roller, etc.)	Impact rates for the individual bearing components. Also, vibrations at very high frequencies (20 to 60 kHz).	Radial	Uneven vibration levels, often with shocks.
JOURNAL BEARINGS LOOSE IN HOUSINGS	Sub-harmonics of shaft rpm; exactly 1/2 or 1/3X rpm.	Primarily Radial	Looseness may only develop at operating speed and temperature (eg, turbomachines).
OIL FILM WHIRL OR WHIP IN JOURNAL BEARINGS	Slightly less than 1/2 shaft speed (42% to 48%).	Primarily Radial	Applicable to high-speed (eg, turbomachines).
HYSTERESIS WHIRL	Shaft critical speed	Primarily Radial	Vibrations excited when passing through critical shaft speed are maintained at higher shaft speeds. Can sometimes be cured by checking tightness of rotor components.

VIBRATION TROUBLE SHOOTING CHART (cont'd)

	FREQUENCY OF VIBRATION (Hz = rpm/60)	DIRECTION	REMARKS
DAMAGED OR WORN GEARS	Tooth meshing frequencies (shaft rpm x no. of teeth and harmonics)	Radial and Axial	Sidebands around tooth meshing frequencies indicate modulation (eg, eccentricity) at frequency corresponding to sideband spacings. Normally only detectable with very narrow-bank analysis.
MECHANICAL LOOSENESS	2X rpm	Radial and Axial	Also sub- and inter-harmonics and for loose journal bearings.
FAULTY BELT DRIVE	1X, 2X, 3X & 4X rpm of belt	Radial	
UNBALANCED RECIPROCATING FORCES & COUPLES	1X rpm and/or multiples for higher order unbalance.	Primarily Radial	
INCREASED TURBULENCE	Blade & vane passing frequencies and harmonics.	Radial and Axial	Increasing levels indicate increasing turbulence.
ELECTRICALLY INDUCED VIBRATIONS	1X rpm or 1X or 2X synchronous frequency	Radial and Axial	Should disappear when turning off the power.
SHAFT POSITION	All	Radial and Axial	Indicates bearing load changes, external forces, process upsets.

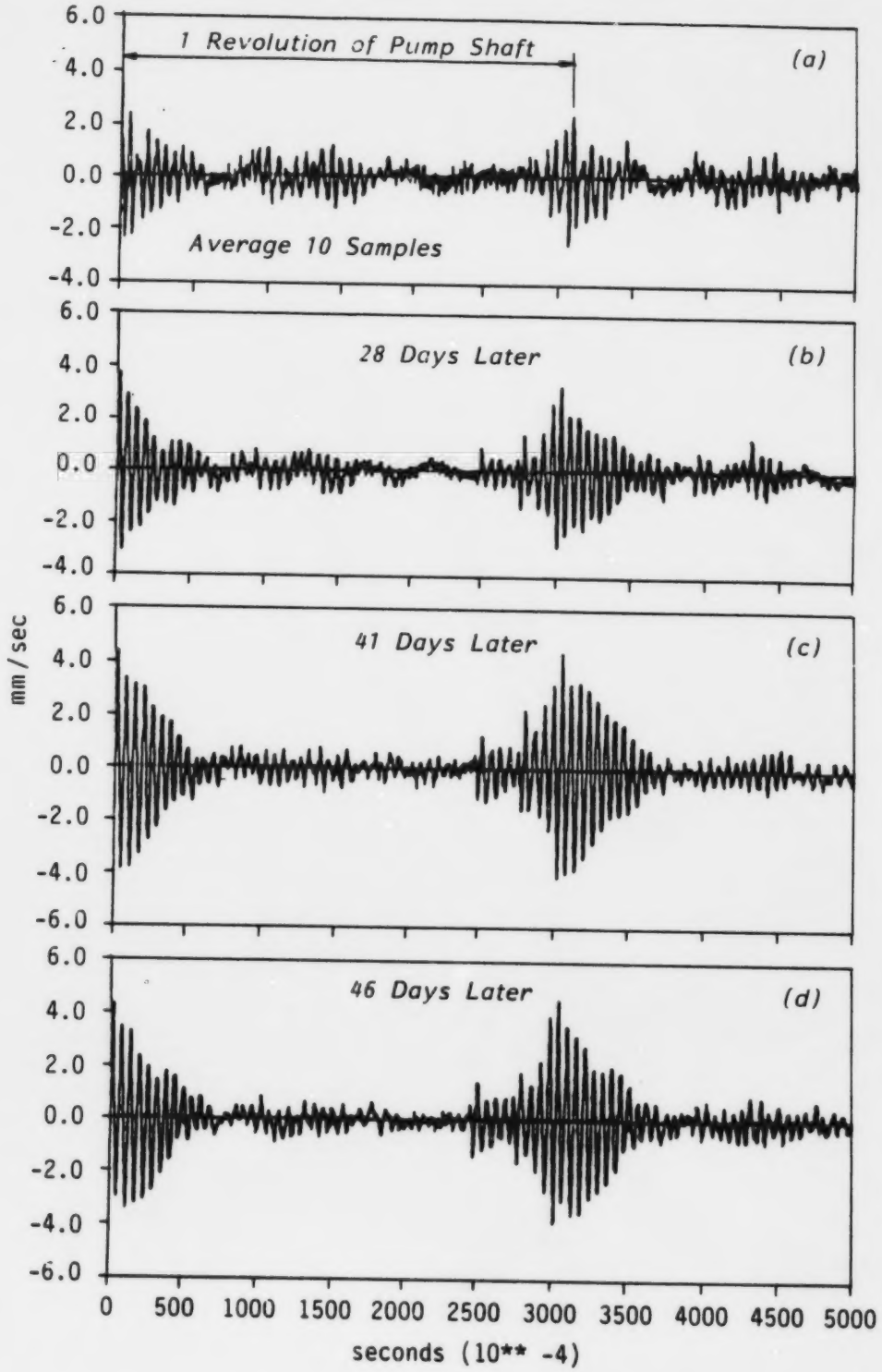


FIGURE 1: SYNCHRONIZED TIME AVERAGE

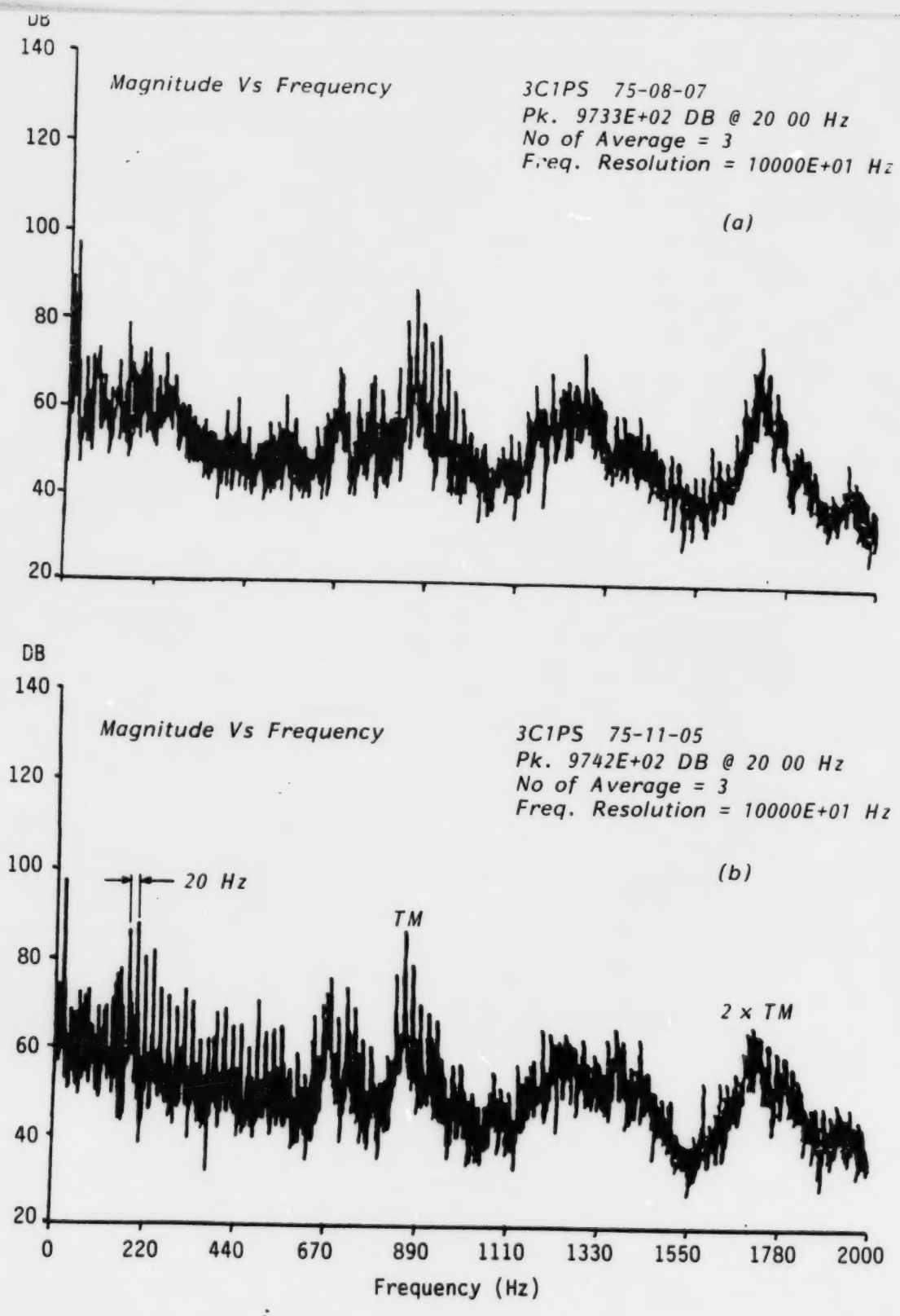


FIGURE 2: FREQUENCY SPECTRA

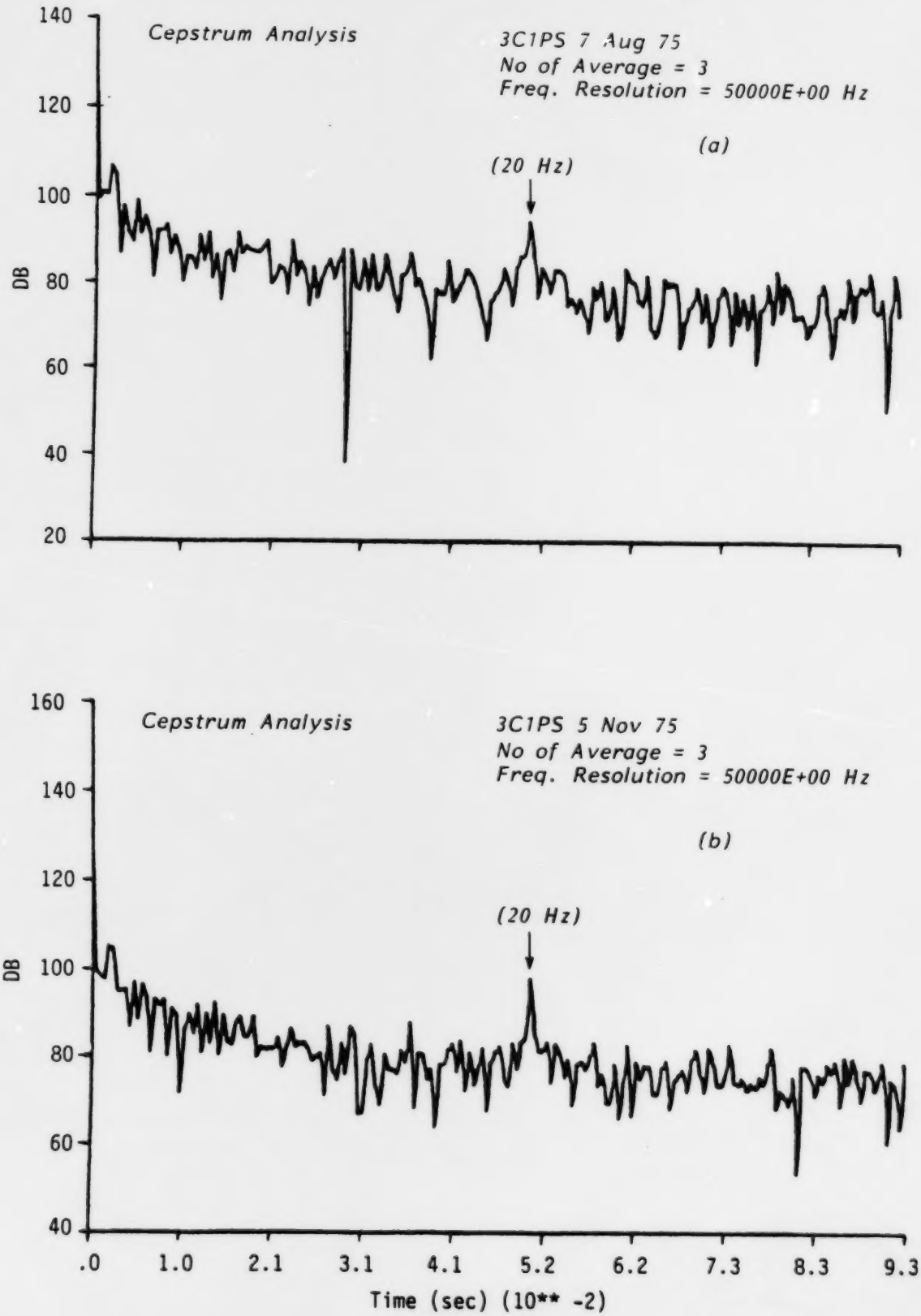


FIGURE 3: CEPSTRUM PLOTS

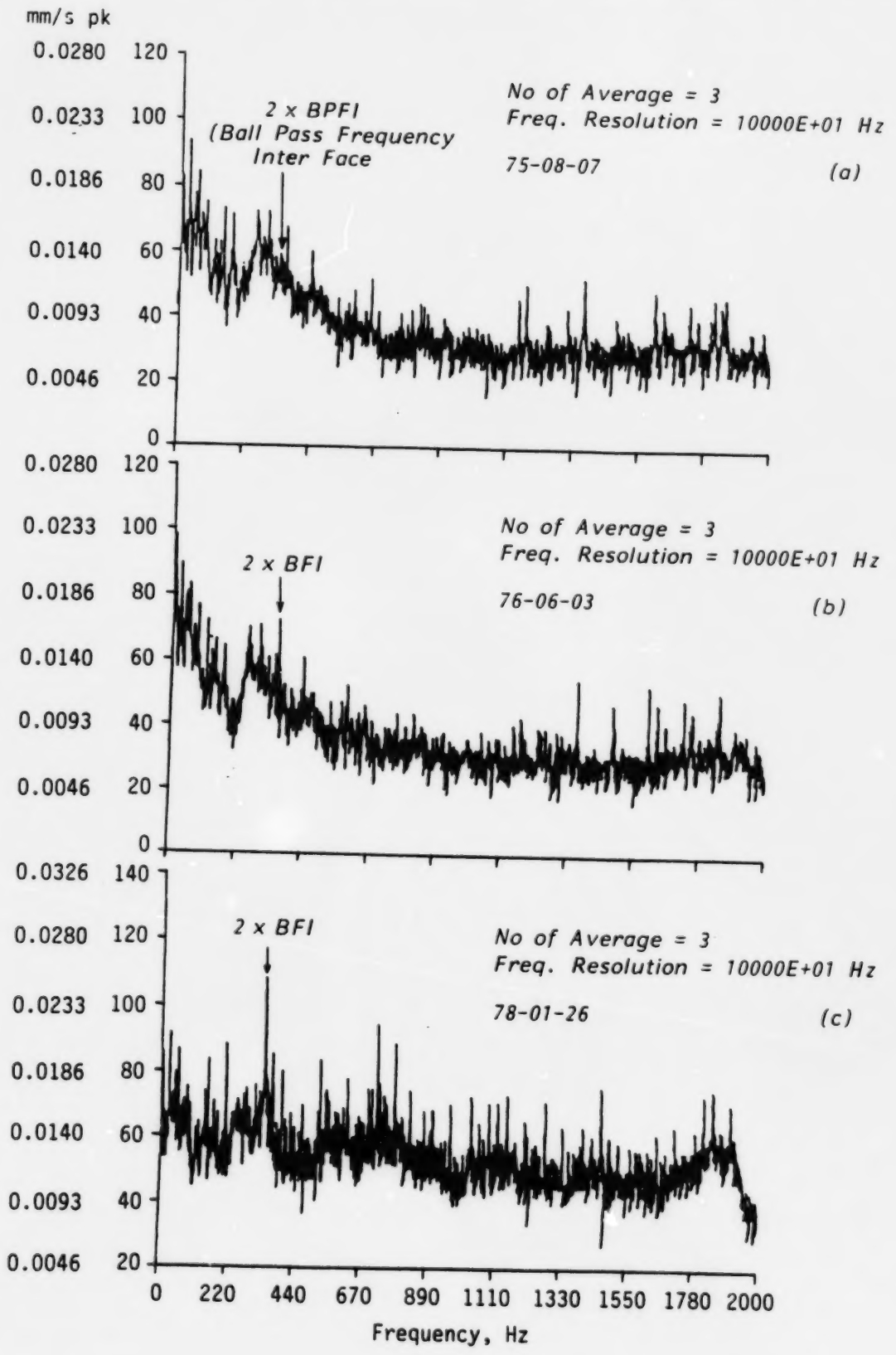


FIGURE 4: FREQUENCY SPECTRA

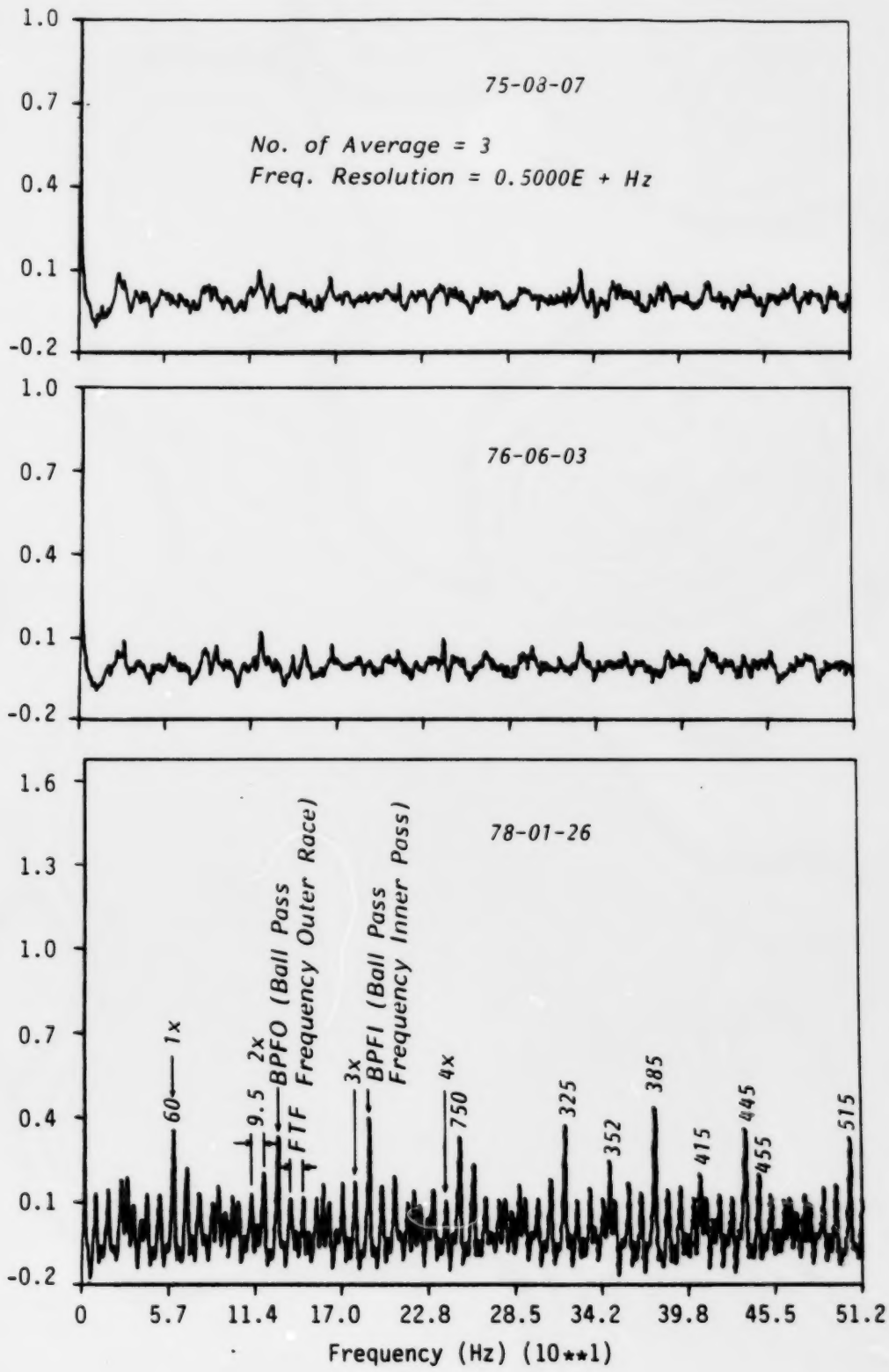


FIGURE 5: NORMALIZED SPECTRAL AUTO CORRELATION

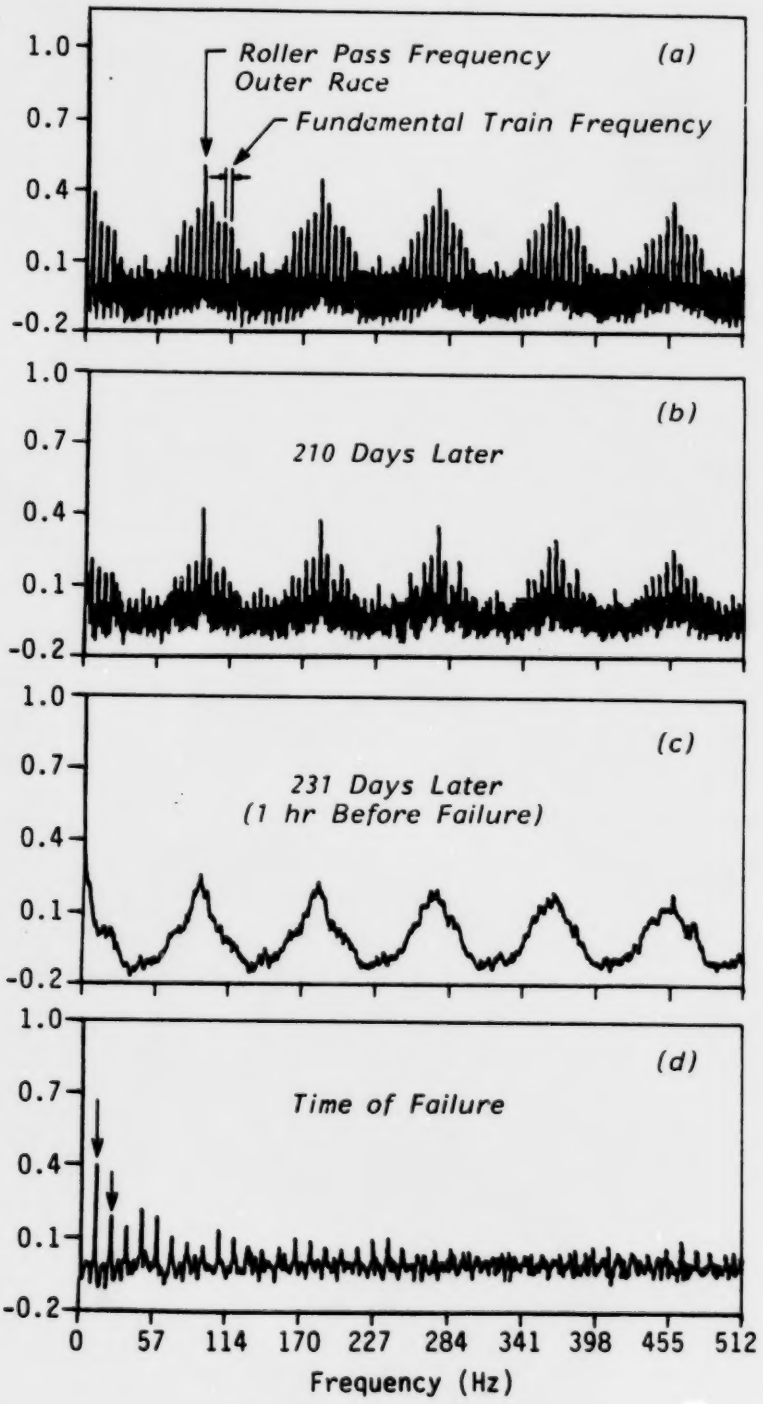


FIGURE 6: NORMALIZED SPECTRAL AUTO CORRELATION

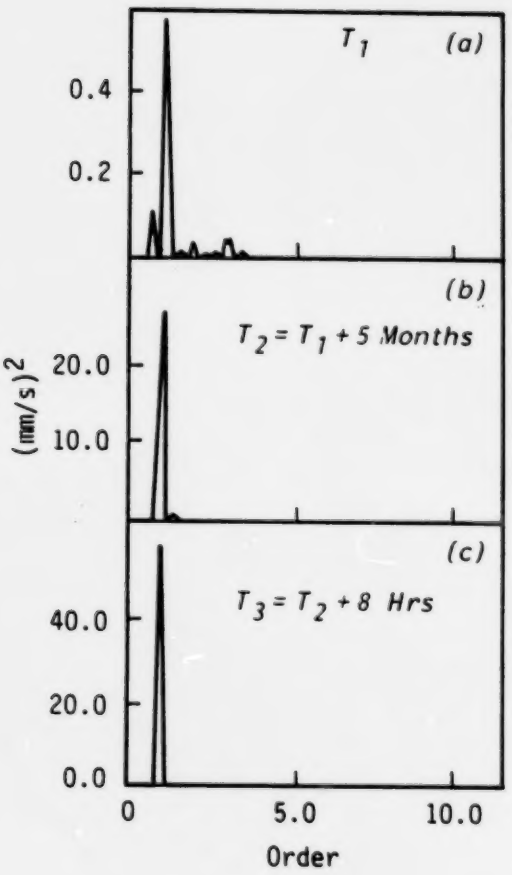


FIGURE 7: DETECTION OF IMPELLER VANE FATIGUE USING ORDER DOMAIN ANALYSIS

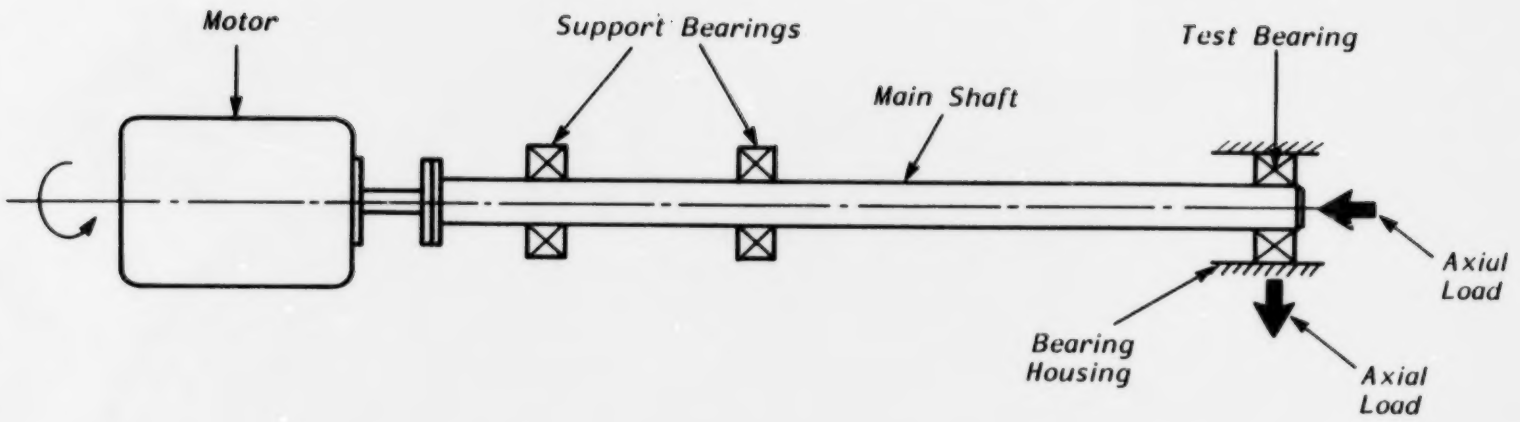


FIGURE 8: SCHEMATIC OF TEST RIG

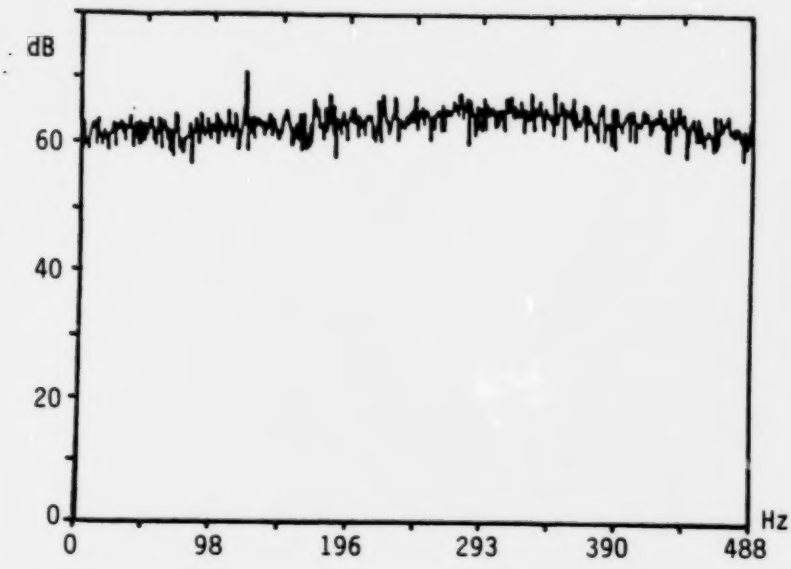
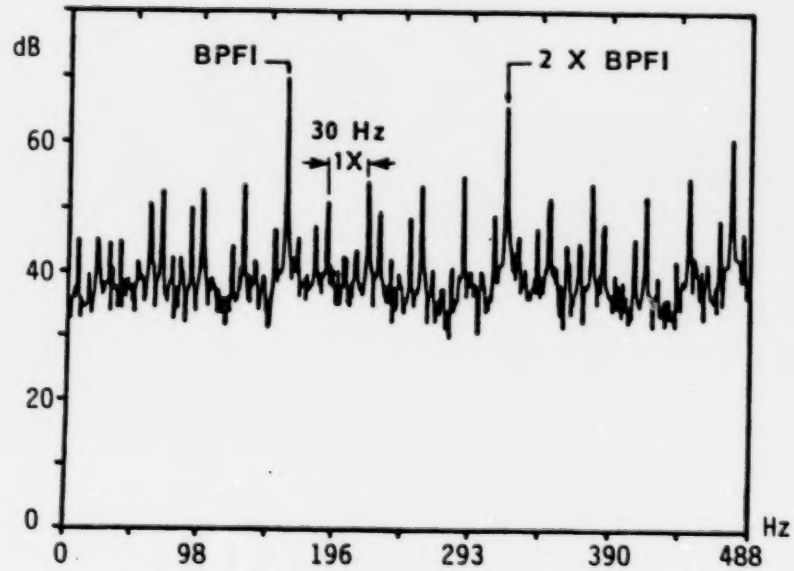


FIGURE 9: FREQUENCY SPECTRUM



BPFI = Ball Pass Frequency - Inner Race

FIGURE 10: OUTPUT OF HFRT ALGORITHM

D - Displacement Transducer
V - Velocity Transducer
A - Accellerometer
P - Phase Marker

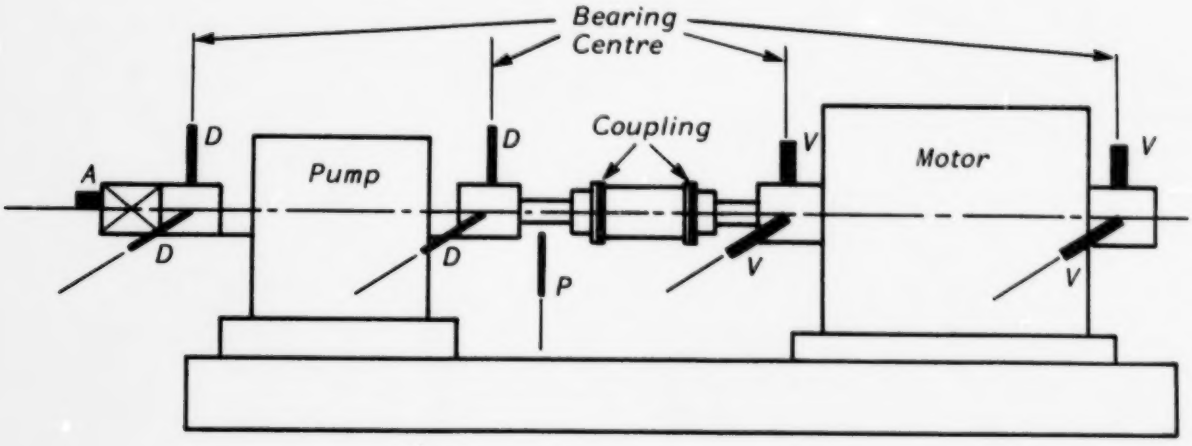
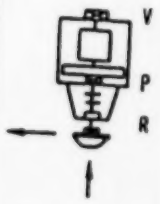
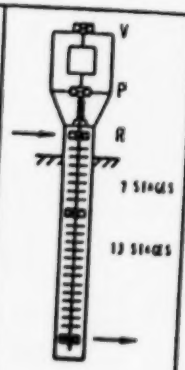
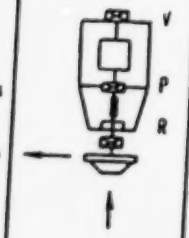


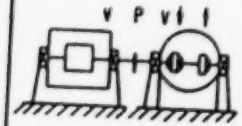
FIGURE 11: INSTRUMENTATION LAY-OUT FOR MAIN MODERATOR PUMP

TRANSDUCERS
V - VELOCITY
R - RUNOUT
P - PHASE

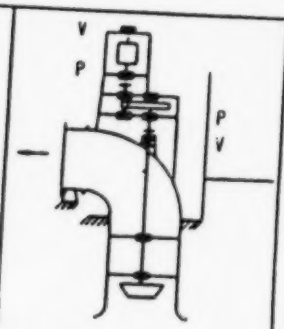


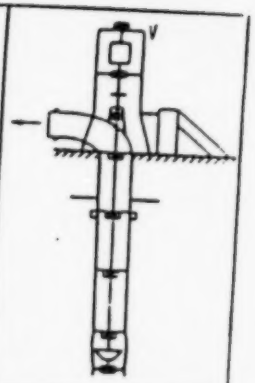






AUXILIARY GENERALLY SIMILAR TO MAIN PUMP





NAME	PRIMARY HEAT TRANSPORT CIRCULATING	PRIMARY HEAT TRANSPORT PRESSURIZING	MODERATOR CIRCULATING	MAIN BOILER FEED	AUXILIARY BOILER FEED	CONDENSER COOLING WATER	LOW PRESSURE SERVICE WATER
HORSEPOWER	1800	400	250	3500	350	1200	1020
RPM	1791	3582	1791	3582	3582	1194,201	717
WATER	HEAVY	HEAVY	HEAVY	LIGHT	LIGHT	LIGHT	LIGHT
NO PER UNIT	16	2	5	3	1	2	3
REDUNDANCY	1/4	1/2	1/5	1/3	0/1	0/2	1/3
CAPACITY, PERCENT	8.3	100	25	50	5	80	70
INTER-PUMP COUPLING	CLOSE	MEDIUM	CLOSE	SMALL	NA	VERY SMALL	SMALL
SUCTION PRESSURE, psia	1250	20	15	38	-	9	9
SUCTION TEMPERATURE °C	250	38	40	130	-	15	15
NO OF STAGES	1	20	1	2	-	1	1
NO OF ROTOR VANES	5	5	5	5,7	-	4	4
NO OF DIFFUSER VANES	2	-	2	2	-	-	7

FIGURE 12: 'PICKERING NGS 'A' - PUMPSET ARRANGEMENT AND SPECIFICATIONS

170596 RD

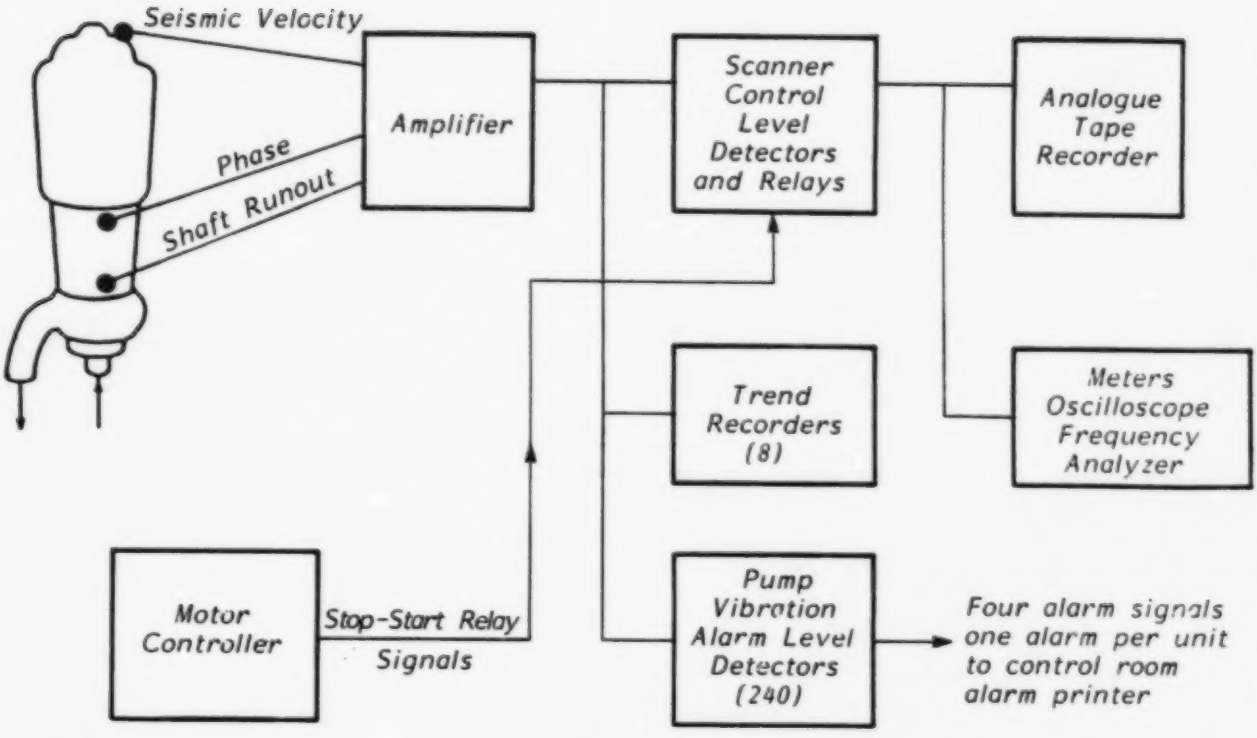


FIGURE 13: VIBRATION DATA COLLECTION SYSTEM AT PICKERING NGS 'A'

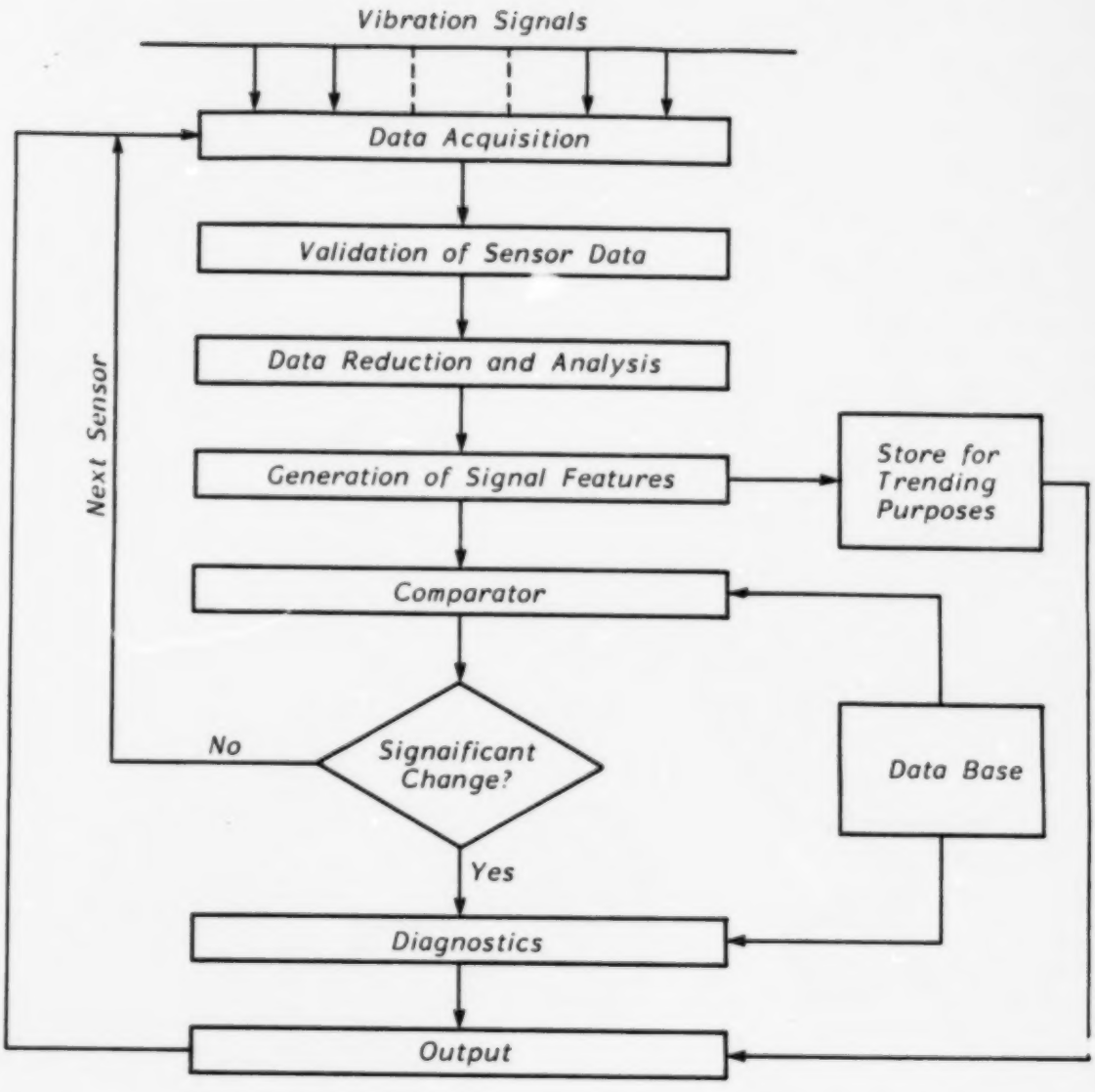


FIGURE 14: ARCHITECTURE OF CONTINUOUS MONITORING SYSTEM

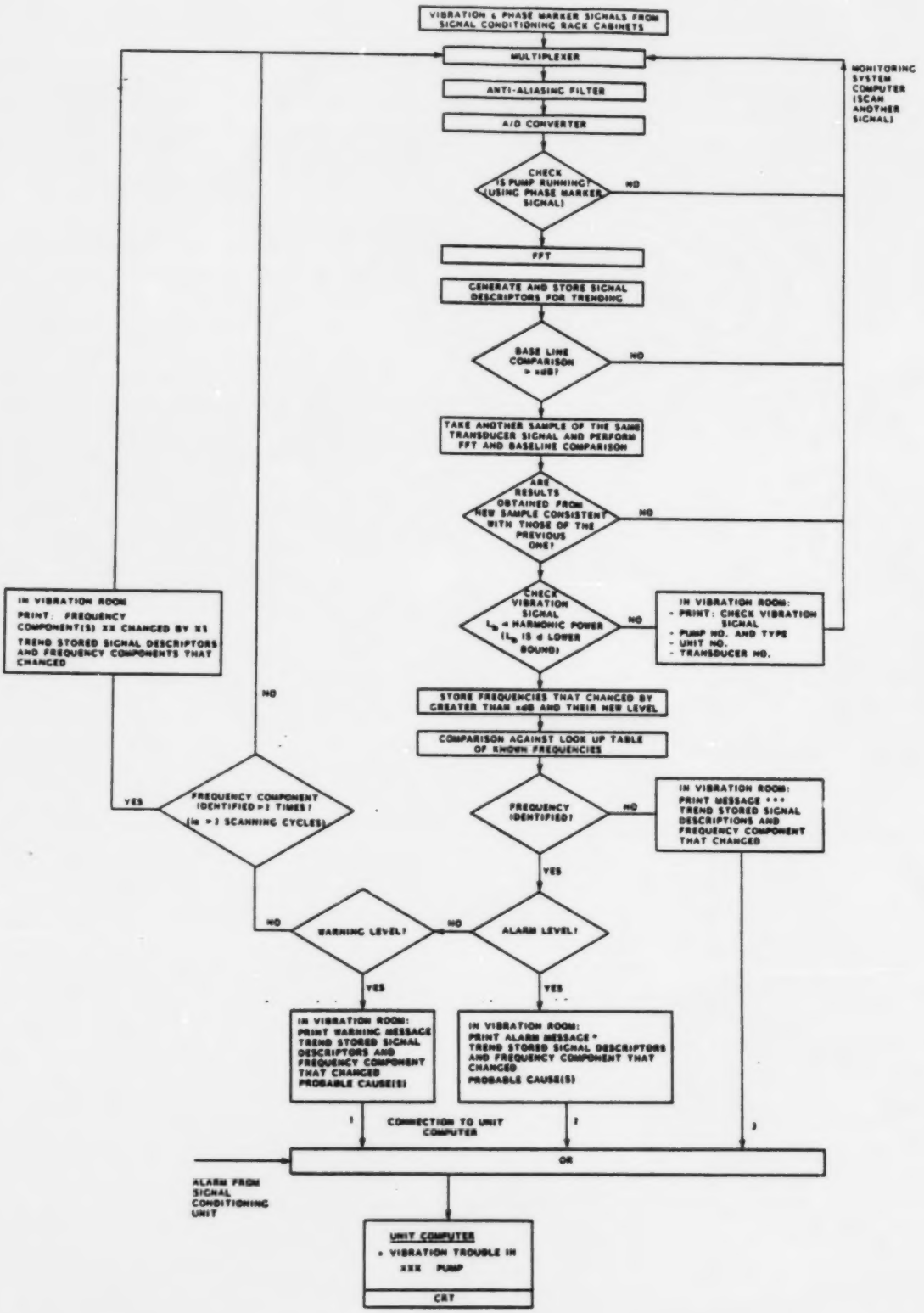

VIBRATION & PHASE MARKER SIGNALS FROM SIGNAL CONDITIONING RACK CABINETS
MULTIPLEXER
ANTI-ALIASING FILTER
A/D CONVERTER
CHECK (IS PUMP RUNNING? (USING PHASE MARKER SIGNAL)
NO
MONITORING SYSTEM COMPUTER (SCAN ANOTHER SIGNAL)
FFT
GENERATE AND STORE SIGNAL DESCRIPTORS FOR TRENDING
BASE LINE COMPARISON > xdB?
NO
TAKE ANOTHER SAMPLE OF THE SAME TRANSDUCER SIGNAL AND PERFORM FFT AND BASELINE COMPARISON
ARE RESULTS OBTAINED FROM NEW SAMPLE CONSISTENT WITH THOSE OF THE PREVIOUS ONE?
NO
CHECK VIBRATION SIGNAL L_b < HARMONIC POWER (L_b IS a LOWER BOUND)
NO
IN VIBRATION ROOM:
- PRINT: CHECK VIBRATION SIGNAL
- PUMP NO. AND TYPE
- UNIT NO.
- TRANSDUCER NO.
IN VIBRATION ROOM
PRINT: FREQUENCY COMPONENT(S) XX CHANGED BY X1
TREND STORED SIGNAL DESCRIPTORS AND FREQUENCY COMPONENTS THAT CHANGED
STORE FREQUENCIES THAT CHANGED BY GREATER THAN xdB AND THEIR NEW LEVEL
COMPARISON AGAINST LOOK UP TABLE OF KNOWN FREQUENCIES
NO
YES
FREQUENCY COMPONENT IDENTIFIED > J TIMES? (in > J SCANNING CYCLES)
FREQUENCY IDENTIFIED?
NO
IN VIBRATION ROOM:
PRINT MESSAGE ***
TREND STORED SIGNAL DESCRIPTIONS AND FREQUENCY COMPONENT THAT CHANGED
YES
NO
WARNING LEVEL?
NO
ALARM LEVEL?
YES
YES
IN VIBRATION ROOM:
PRINT WARNING MESSAGE
TREND STORED SIGNAL DESCRIPTORS AND FREQUENCY COMPONENT THAT CHANGED
PROBABLE CAUSE(S)
IN VIBRATION ROOM:
PRINT ALARM MESSAGE *
TREND STORED SIGNAL DESCRIPTORS AND FREQUENCY COMPONENT THAT CHANGED
PROBABLE CAUSE(S)
1
CONNECTION TO UNIT COMPUTER
2
3
OR
ALARM FROM SIGNAL CONDITIONING UNIT
UNIT COMPUTER
• VIBRATION TROUBLE IN XXX PUMP
CRT

FIGURE 15: FLOW DIAGRAM OF SYSTEM OPERATION

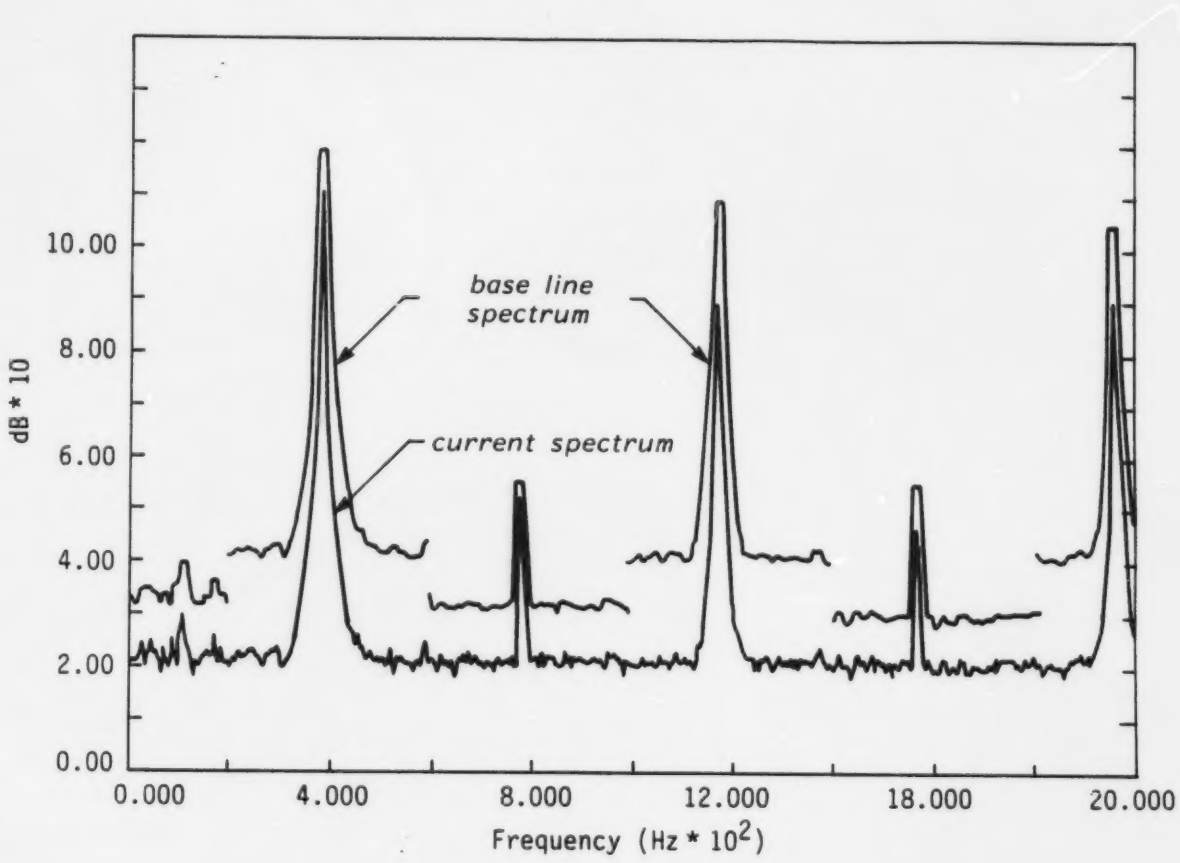
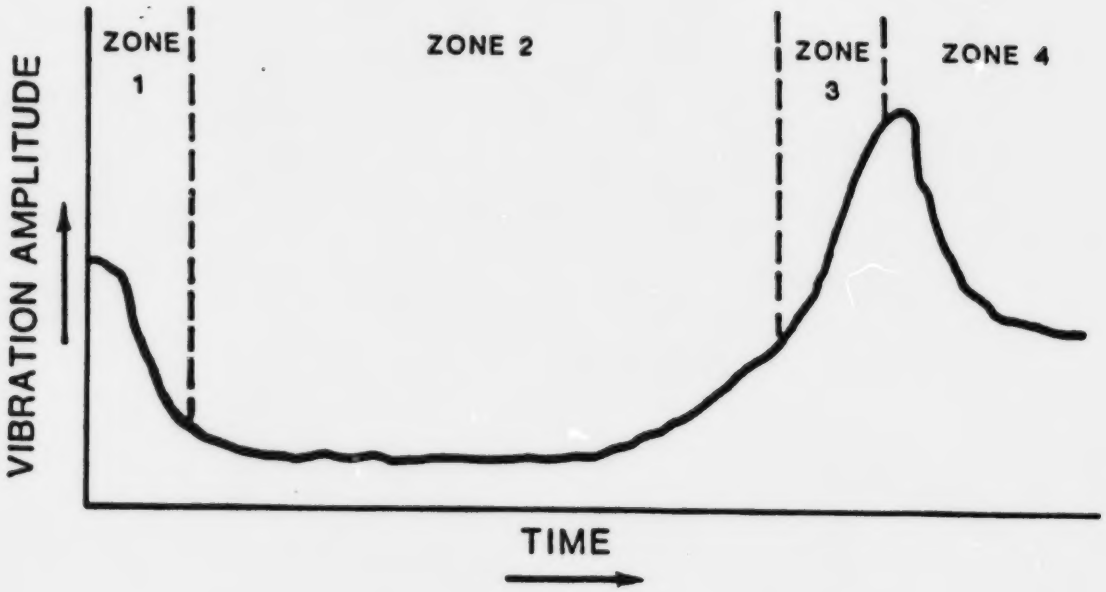


FIGURE 16: BASE LINE COMPARISON



ZONE 1 – New Machine Roughness

ZONE 2 – Useful Machine Life

ZONE 3 – Corrective Action Should Be Taken

ZONE 4 – Component Failure With Either Continuing Increase Or Partial Reduction Of Vibration

FIGURE 17: EXAMPLE OF VIBRATION TREND CURVE /10/

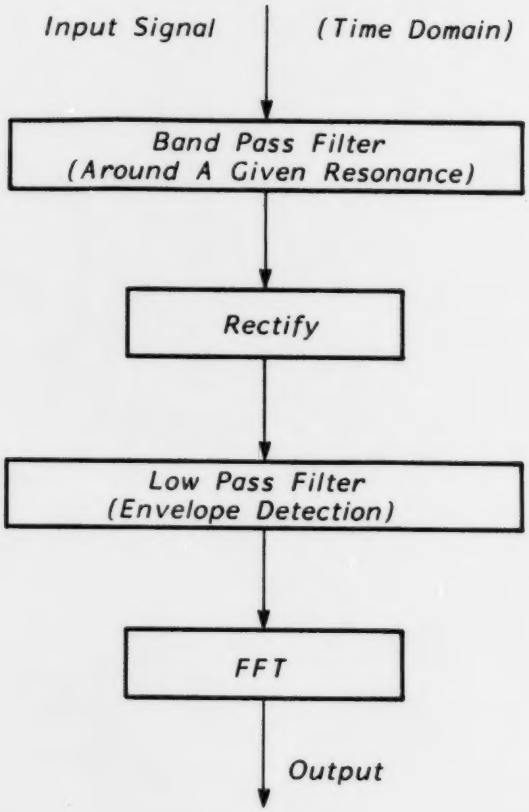


FIGURE 18: FLOWCHART OF THE HFRT ALGORITHM

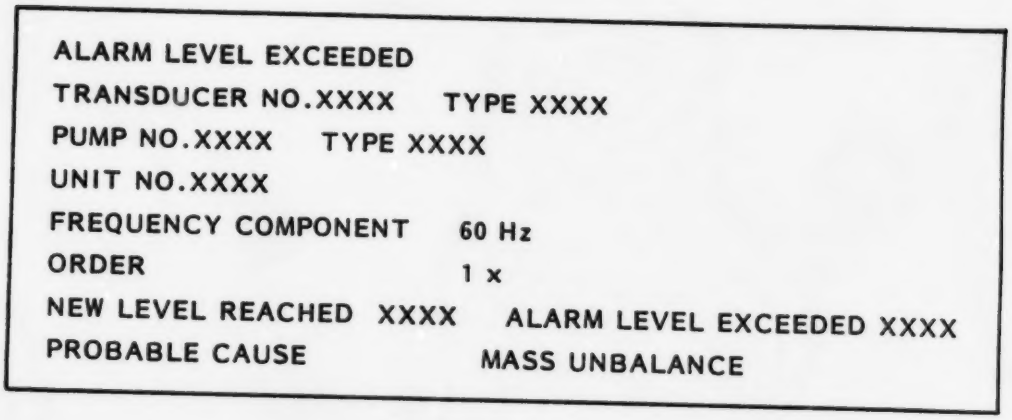

ALARM LEVEL EXCEEDED
TRANSDUCER NO.XXXX TYPE XXXX
PUMP NO.XXXX TYPE XXXX
UNIT NO.XXXX
FREQUENCY COMPONENT 60 Hz
ORDER 1 x
NEW LEVEL REACHED XXXX ALARM LEVEL EXCEEDED XXXX
PROBABLE CAUSE MASS UNBALANCE

FIGURE 19: EXAMPLE OF A DIAGNOSTIC MESSAGE

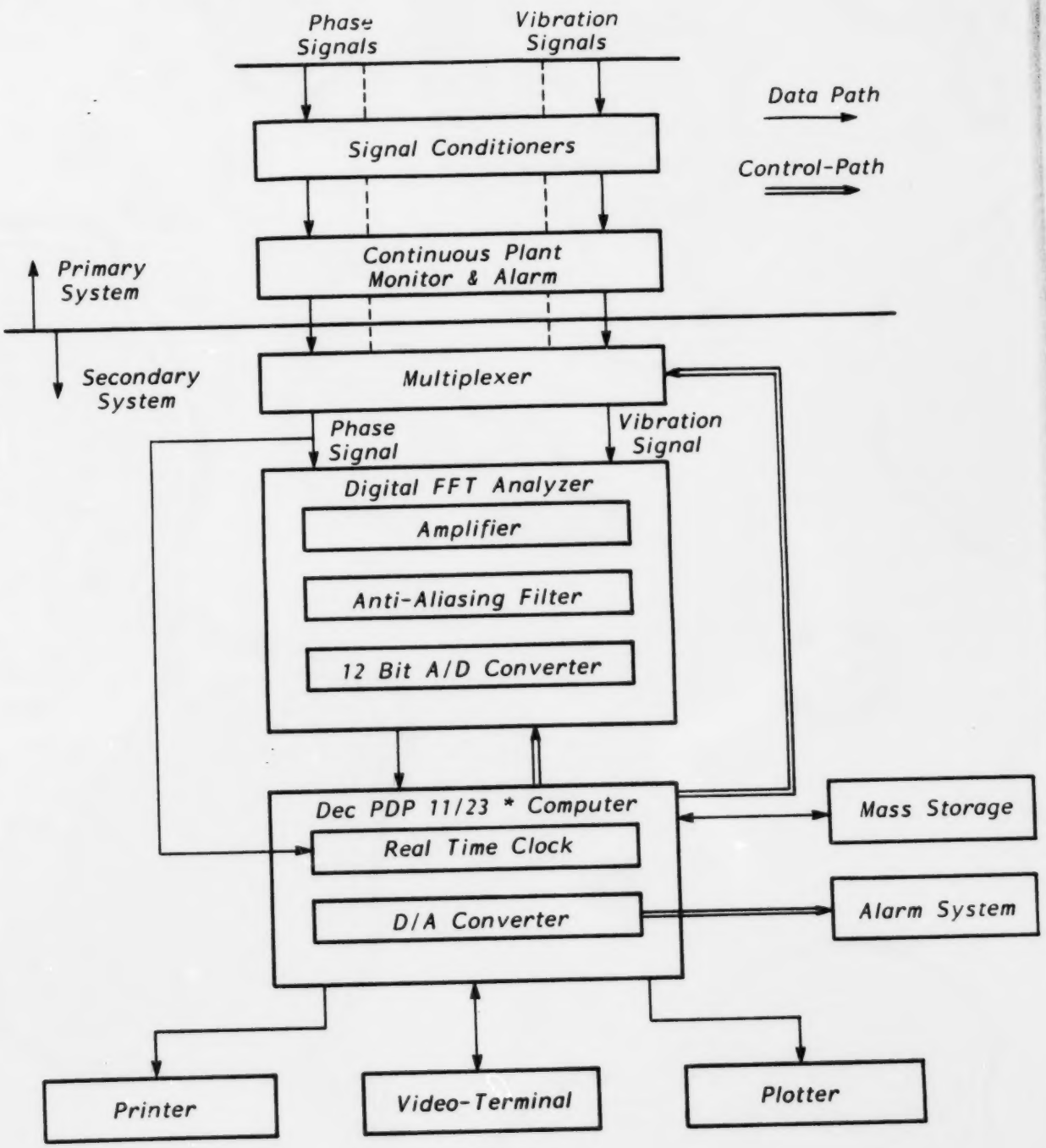


FIGURE 20: HARDWARE CONFIGURATION

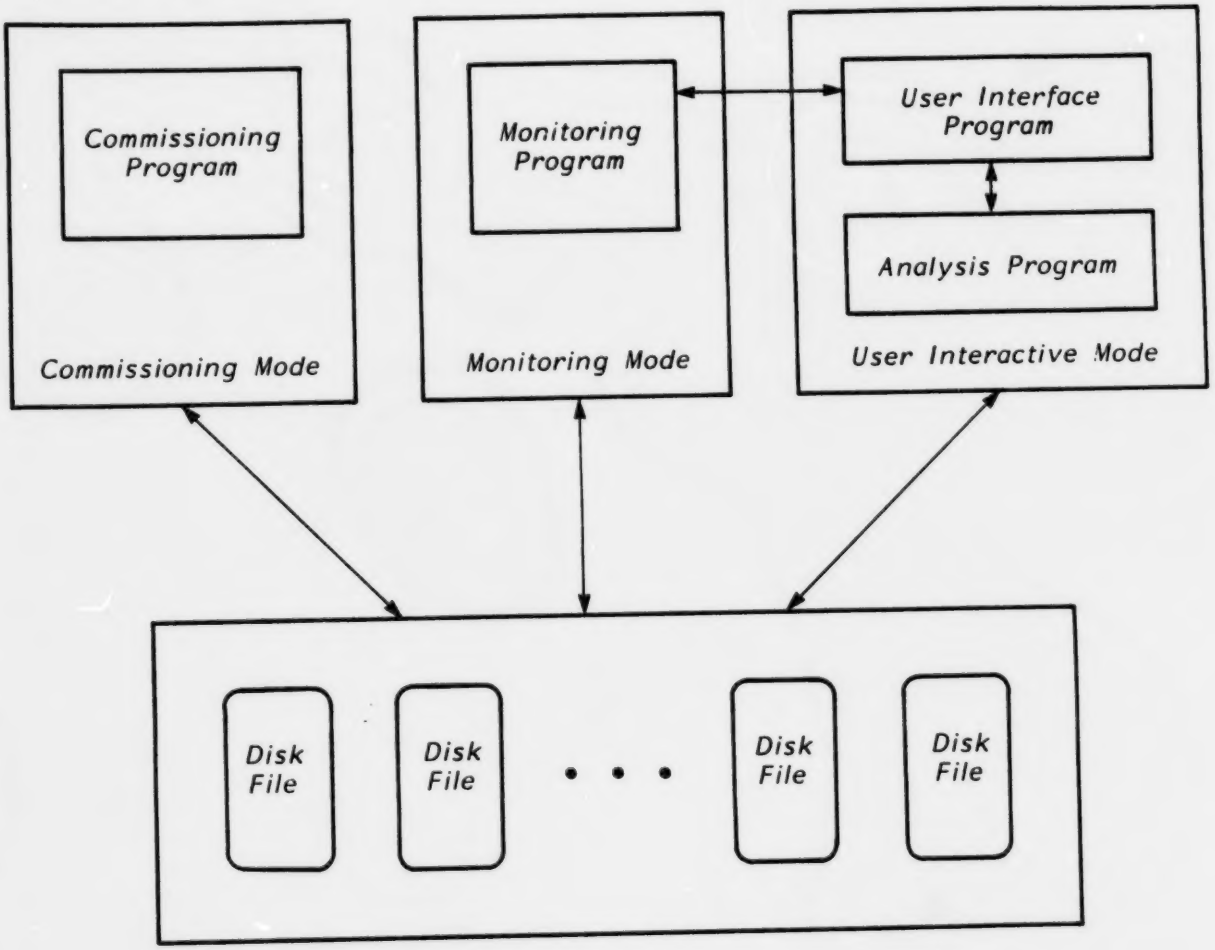
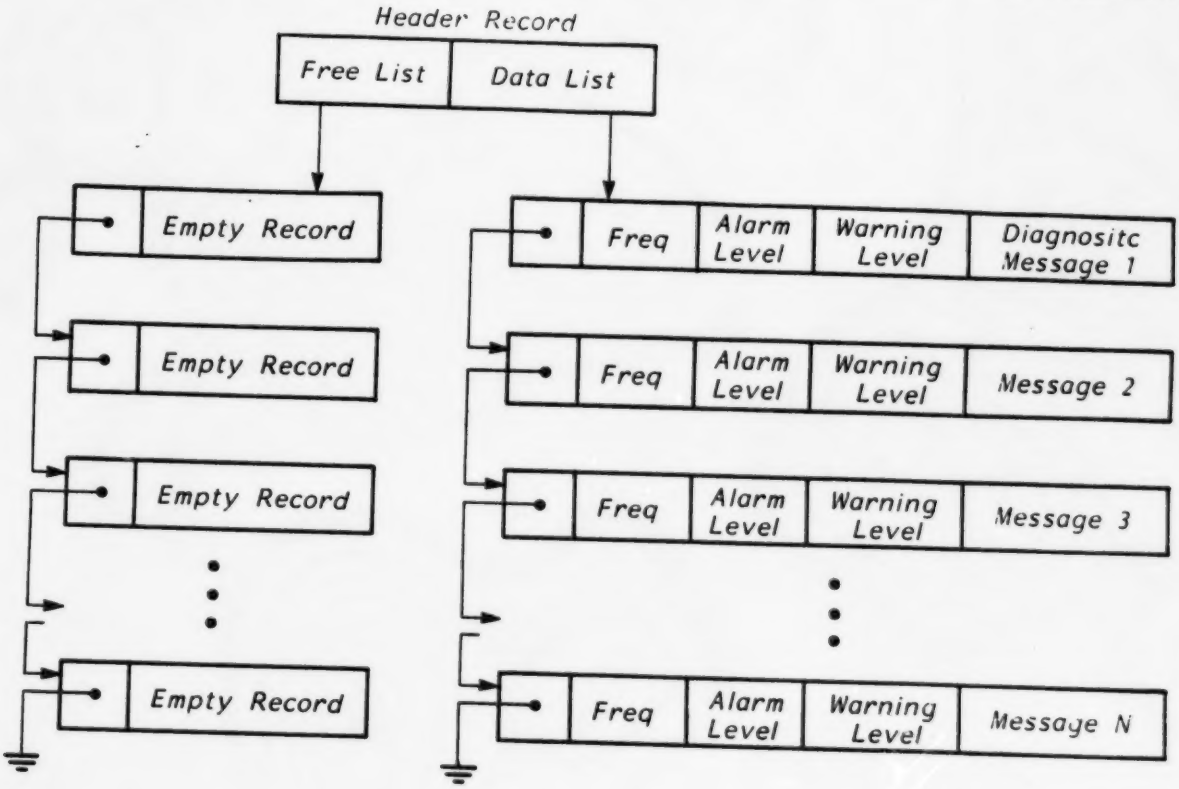
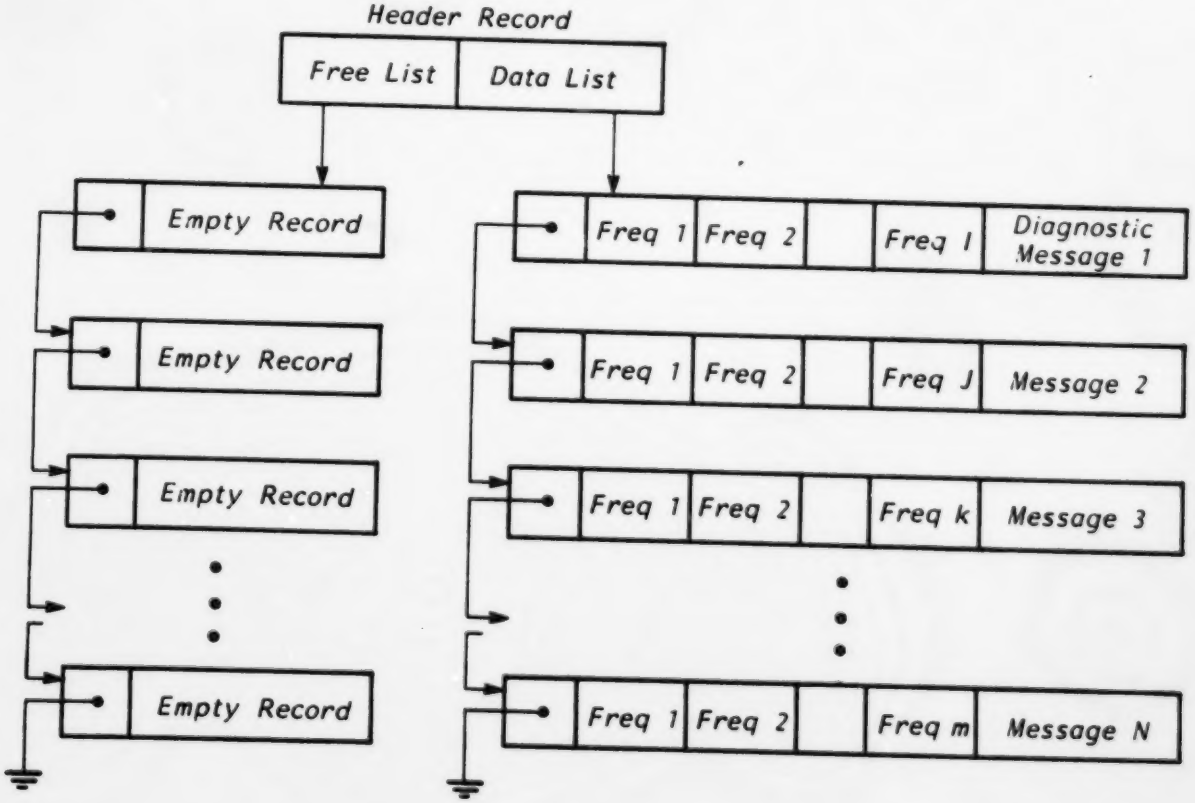


FIGURE 21: SOFTWARE CONFIGURATION



A) For Single Frequency (Per Transducer)



B) For Multiple Frequencies (Per Pump)

FIGURE 22: LINKED LISTS FOR FREQUENCY DIAGNOSTICS

APPENDIX A

INSTRUMENTATION LAYOUT ON PUMPSETS

CONDENSATE EXTRACTION PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 3
SPEED = 1185 RPM
TYPE = VERTICAL
TOTAL WEIGHT = 11,220 KG

MOTOR

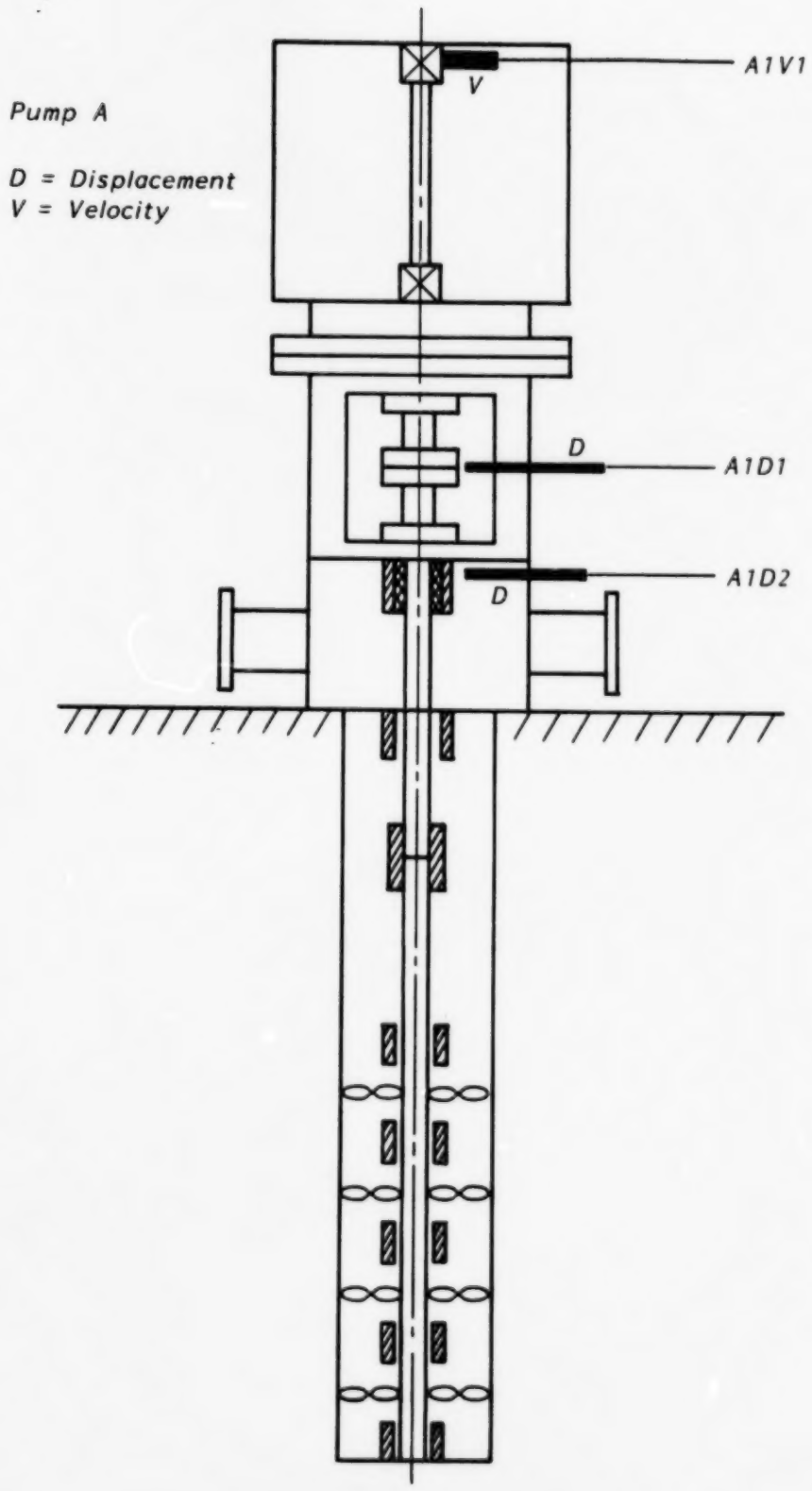
MANUFACTURER = CANADIAN GENERAL ELECTRIC
POWER = 1,325 KW PH/HZ/V = 3/60/4000
MOTOR WEIGHT = 5,760 KG
TYPE = INDUCTION
BEARING TYPE = ANTIFRICTION

PUMP

MANUFACTURER = BYRON JACKSON
PUMP SIZE AND TYPE = 28 $CKXF_1H_3$ 4 STG VMT
SEAL SIZE AND TYPE = BW V-4250
BEARING TYPE = SLEEVE, WATER LUBRICATED
NUMBER OF BLADES =
NUMBER OF STAGES = 4
PUMP WEIGHT = 5,460 KG
CAPACITY =
DESIGN HEAD =
COUPLING = RIGID

TRANSDUCERS

1 - PHASE
1 - DISPLACEMENT ABOVE PUMP SEAL
1 - VELOCITY ON MOTOR (UPPER BEARING)



CONDENSATE EXTRACTION PUMPSETS (3 PUMPS/UNIT)

170607 RD

AUXILIARY CONDENSATE EXTRACTION PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 1
SPEED =
TYPE = VERTICAL
TOTAL WEIGHT =

MOTOR

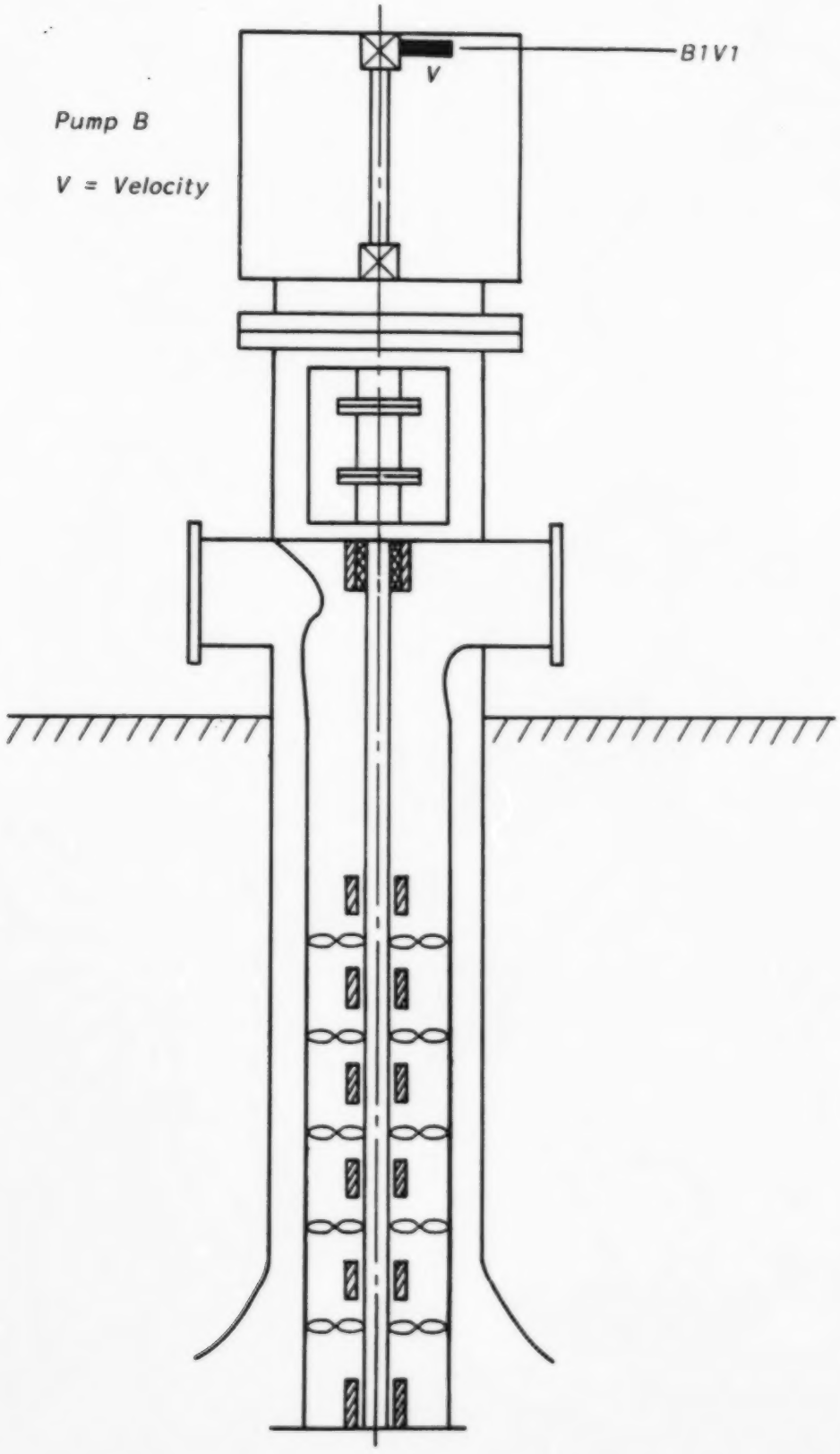
MANUFACTURER = BROWN BOVERI CANADA
POWER =
MOTOR WEIGHT =
PH/HZ/V =
TYPE =
BEARING TYPE = ANTIFRICTION

PUMP

MANUFACTURER = CANADA MACHINERY CORPORATION
PUMP SIZE AND TYPE =
SEAL SIZE AND TYPE =
BEARING TYPE = SLEEVE, WATER LUBRICATED
NUMBER OF BLADES =
NUMBER OF STAGES = 5
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING = RIGID

TRANSDUCERS

1 - VELOCITY ON MOTOR (UPPER BEARING)



AUXILIARY CONDENSATE EXTRACTION PUMPSETS (1 PUMP/UNIT)

170608 RD

MAIN BOILER FEED PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 4
SPEED = 3,573 RPM
TYPE = HORIZONTAL
TOTAL WEIGHT = 18,824 KG

MOTOR

MANUFACTURER = CANADIAN GENERAL ELECTRIC
POWER = 3,600 KW PH/HZ/V = 3/60/4000
MOTOR WEIGHT = 12,250 KG
TYPE = INDUCTION
BEARING TYPE = JOURNAL, OIL LUBRICATED

PUMP

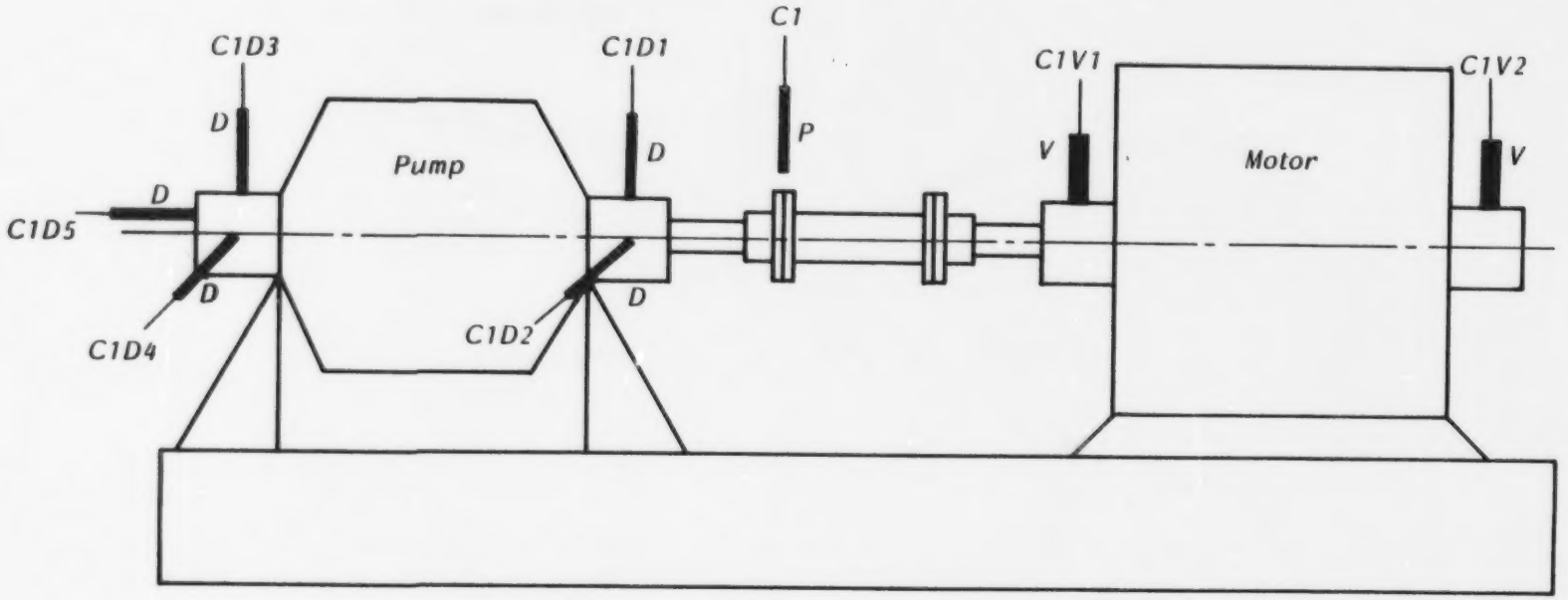
MANUFACTURER = BINGHAM-WILLAMETTE COMPANY
PUMP SIZE AND TYPE = 300 x 300 x 450 B MSD-D
SEAL SIZE AND TYPE = PACKING LESS STUFFING BOX
BEARING TYPE = SLEEVE, OIL LUBRICATED
NUMBER OF BLADES =
NUMBER OF STAGES = 2
PUMP WEIGHT = 3,850 KG
CAPACITY =
DESIGN HEAD =
COUPLING = FLEXIBLE - MM FULLFLEX #5

TRANSDUCERS

1 - PHASE
2 - VELOCITY ON MOTOR INBOARD/OUTBOARD BEARINGS
4 - DISPLACEMENT ON PUMP INBOARD/OUTBOARD BEARINGS
AND PUMP THRUST BEARINGS

Pump C

P = Phase
V = Velocity
D = Displacement



PUMPSET LAYOUT
MAIN BOILER FEED PUMPSETS (4 PUMPS/UNIT)

170609 RD

AUXILIARY BOILER FEED PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 2
SPEED = 3,570 RPM
TYPE = HORIZONTAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER = CANADIAN GENERAL ELECTRIC
POWER = 475 KW PH/HZ/V = 3/60/4000
MOTOR WEIGHT = 1,909 KG
TYPE = INDUCTION
BEARING TYPE = ANTIFRICTION

PUMP

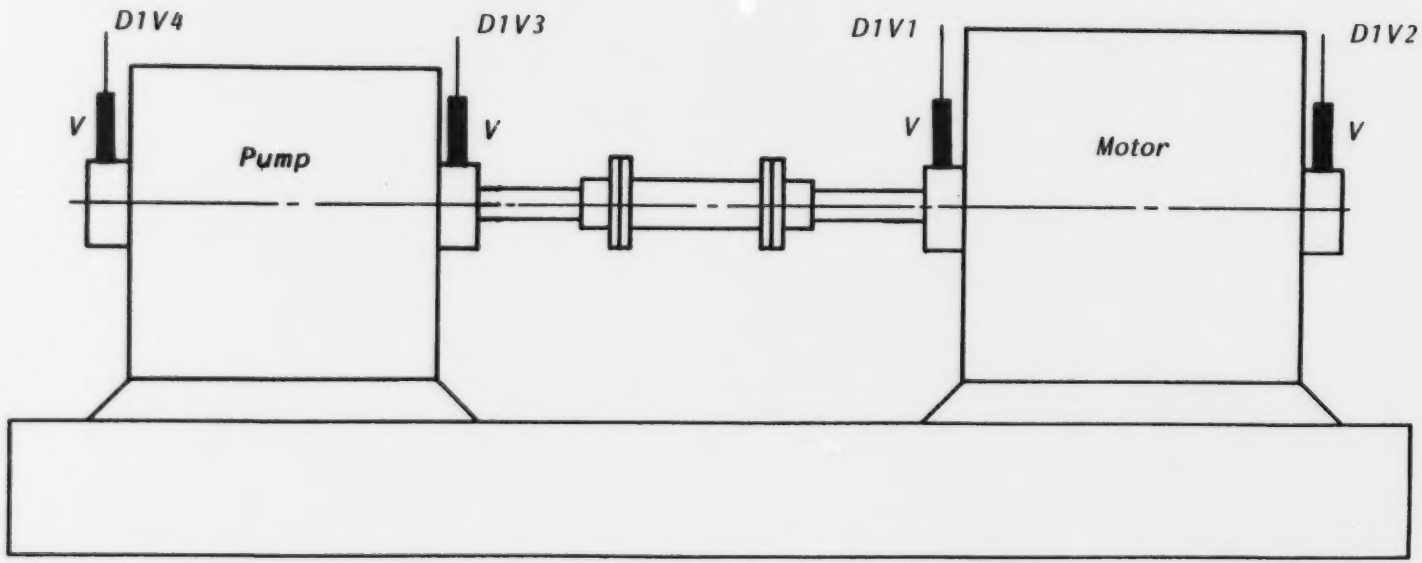
MANUFACTURER =
PUMP SIZE AND TYPE =
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

4 - VELOCITY ON MOTOR INBOARD/OUTBOARD BEARINGS
ON PUMP INBOARD/OUTBOARD BEARINGS

Pump D

V = Velocity



AUXILIARY BOILER FEED PUMPSETS (2/UNIT)

170610 RD

EMERGENCY SERVICE WATER PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 4
SPEED = 1,180 RPM
TYPE = VERTICAL
TOTAL WEIGHT =

MOTOR

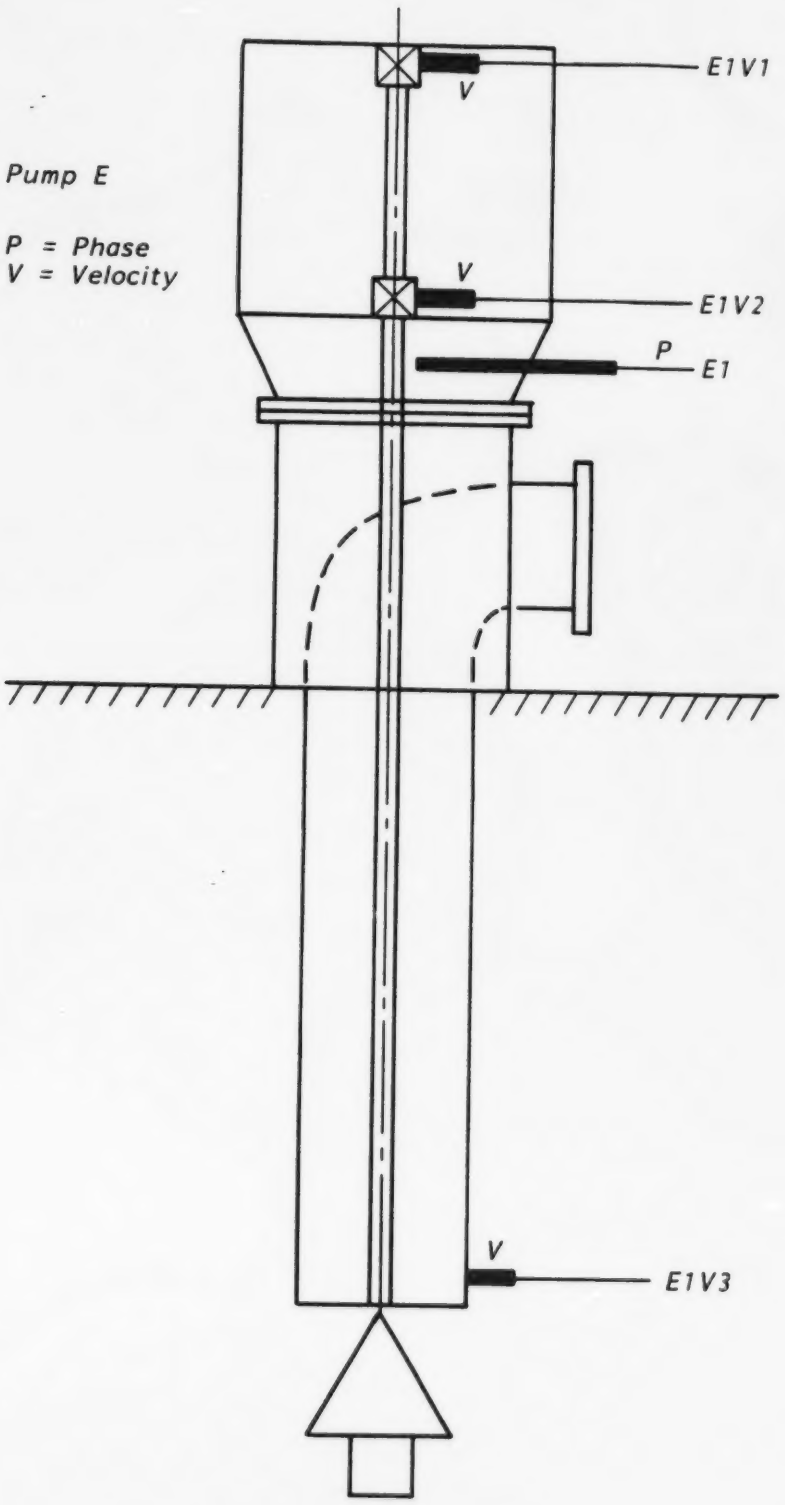
MANUFACTURER = CANADIAN GENERAL ELECTRIC
POWER = 1,000 KW PH/HZ/V = 3/60/4000
MOTOR WEIGHT = 5,602 KG
TYPE = INDUCTION
BEARING TYPE = ANTIFRICTION

PUMP

MANUFACTURER = CANADIAN MACHINERY CORPORATION
PUMP SIZE AND TYPE = 36 APKD 25 TG
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT = 7,348 KG
CAPACITY = 900 L/S
DESIGN HEAD = 82 M
COUPLING =

TRANSDUCERS

1 - PHASE
3 - VELOCITY ON MOTOR TOP/BOTTOM BEARINGS AND ON PUMP BELL



EMERGENCY SERVICE WATER PUMPSETS (4/STATION)

170611 RD

LOW PRESSURE SERVICE WATER PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 4
SPEED = 720 RPM
TYPE = VERTICAL
TOTAL WEIGHT =

MOTOR

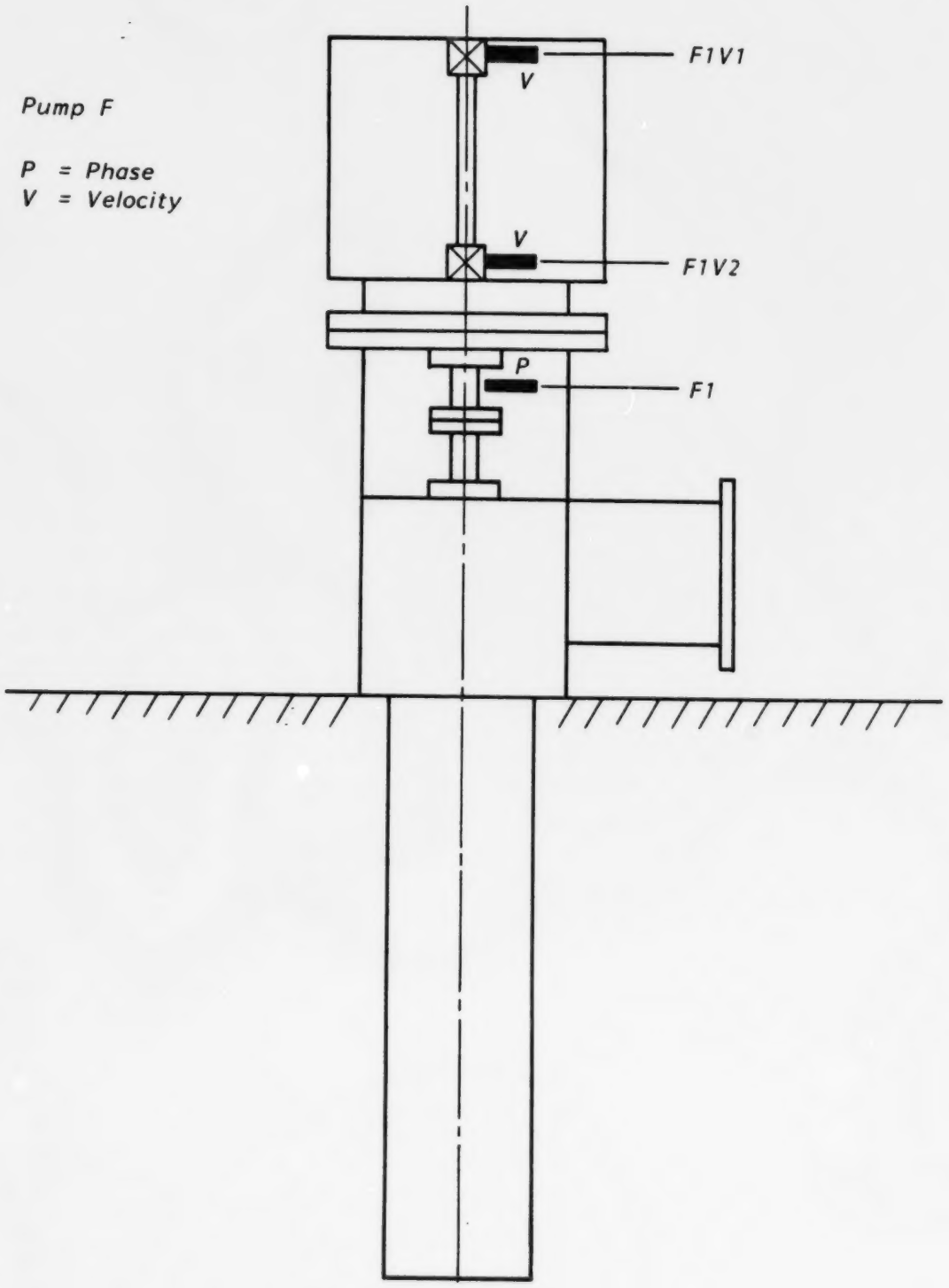
MANUFACTURER = RELIANCE ELECTRIC
POWER = 1,300 KW PH/HZ/V =
MOTOR WEIGHT = 6,800 KG
TYPE =
BEARING TYPE = ANTIFRICTION, TOP BRG SKF 29432E

PUMP

MANUFACTURER = CANADIAN MACHINERY CORPORATION
PUMP SIZE AND TYPE = 45 APH
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT = 11,791 KG
CAPACITY = 2,110 L/S
DESIGN HEAD = 48.2 M
COUPLING =

TRANSDUCERS

1 - PHASE
2 - VELOCITY ON MOTOR TOP/BOTTOM BEARINGS



LOW PRESSURE SERVICE WATER PUMPSET (4/UNIT)

170612 RD

CONDENSER CIRCULATING WATER PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 3
SPEED = 320 RPM
TYPE = VERTICAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER =
POWER =
MOTOR WEIGHT =
TYPE =
BEARING TYPE =

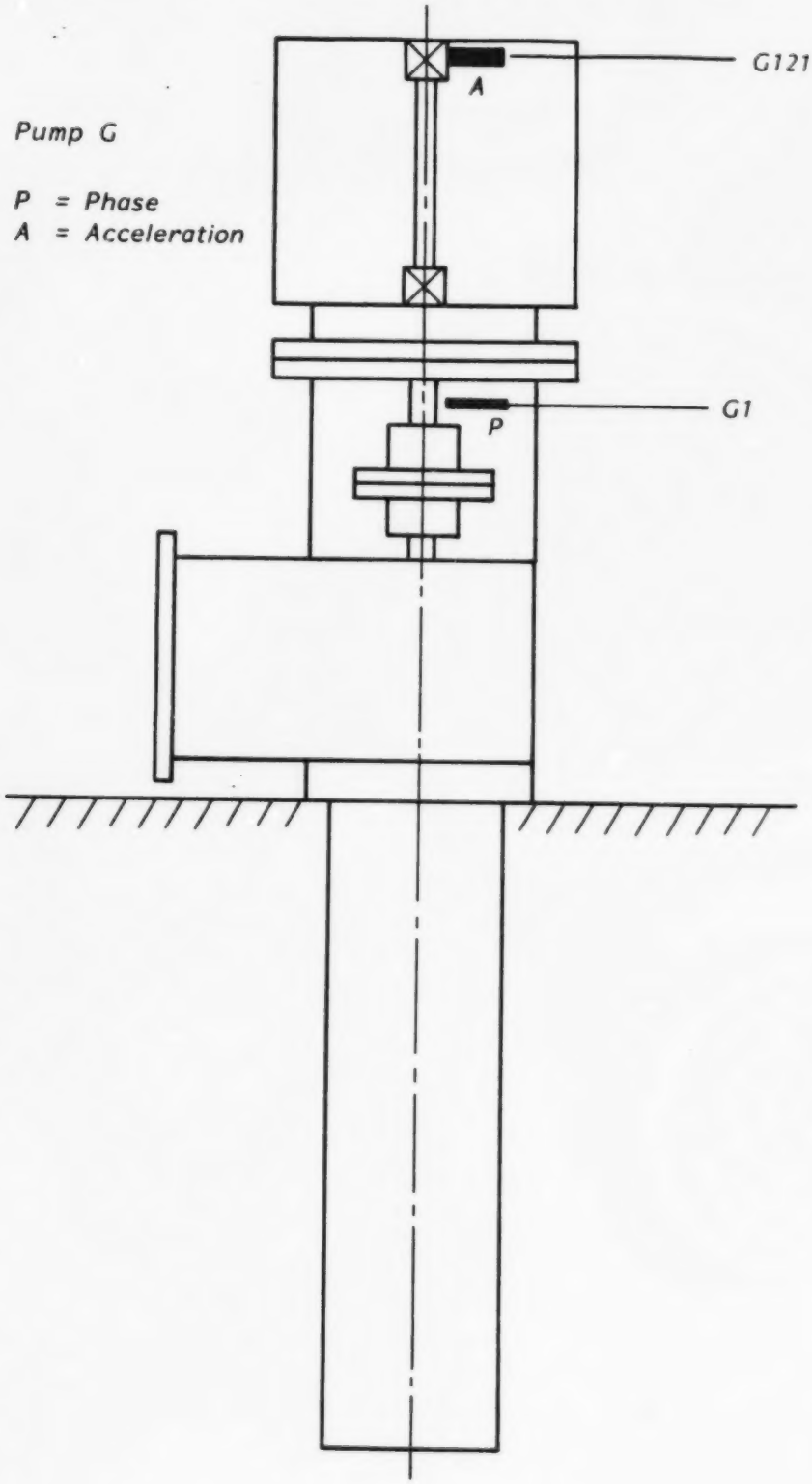
PH/HZ/V =

PUMP

MANUFACTURER = CANADIAN MACHINERY CORPORATION
PUMP SIZE AND TYPE = 69 APMA
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY = 10,536 L/S
DESIGN HEAD = 17 M
COUPLING =

TRANSDUCERS

1 - PHASE
1 - ACCELERATION AT TOP OF MOTOR



CONDENSER CIRCULATING WATER PUMPSETS (3/UNIT)

170613 RD

MAIN MODERATOR PUMPS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 2
SPEED = 3,600 RPM
TYPE = HORIZONTAL
TOTAL WEIGHT = 11,298 KG

MOTOR

MANUFACTURER = RELIANCE ELECTRIC
POWER = 1,000 KW PH/HZ/V = 3/60/400
MOTOR WEIGHT = 5,535 KG
TYPE =
BEARING TYPE = SLEEVE, OIL LUBRICATING

PUMP

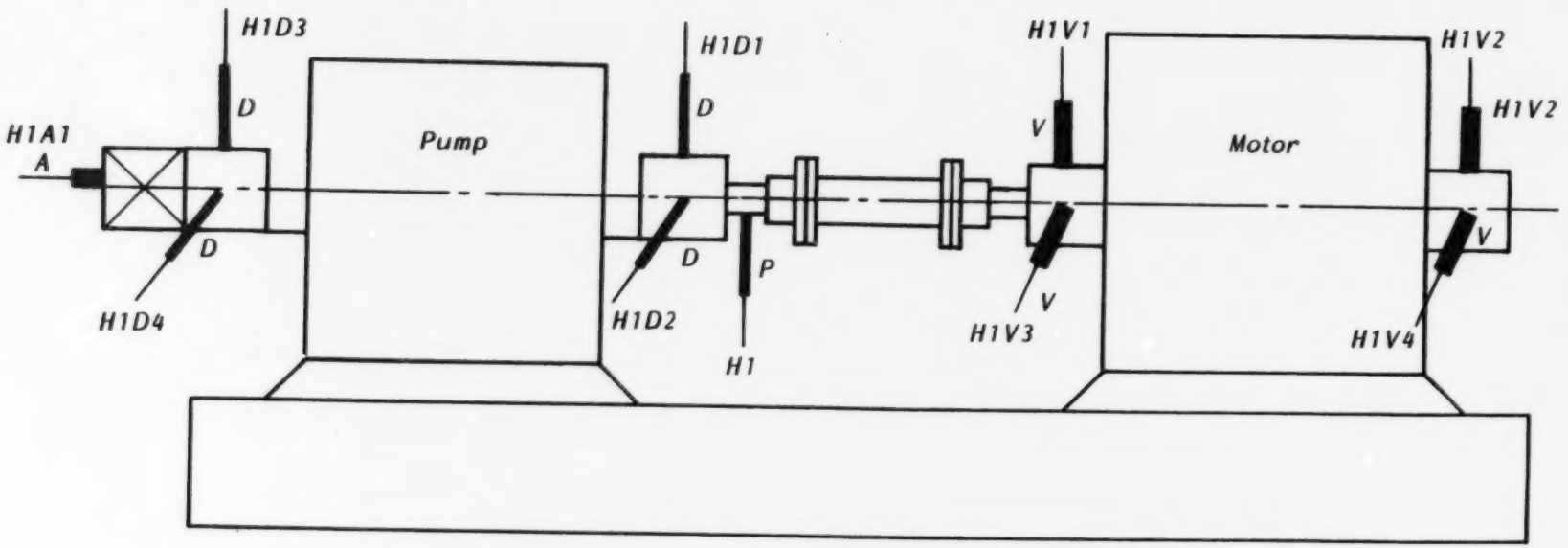
MANUFACTURER = BINGHAM-WILLAMETTE COMPANY
PUMP SIZE AND TYPE = 14 X 16 X 24 CD
SEAL SIZE AND TYPE = MECH SEAL - JOHN CRANE
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES = 1
PUMP WEIGHT = 2,812 KG
CAPACITY =
DESIGN HEAD =
COUPLING = FLEXIBLE, THOMAS SN SIZE 450

TRANSDUCERS

1 - PHASE
4 - DISPLACEMENT AT PUMP INBOARD/OUTBOARD BEARING
4 - VELOCITY AT MOTOR INBOARD/OUTBOARD BEARINGS
1 - ACCELERATION AT PUMP ROLLING ELEMENT THRUST BEARING

Pump H

P = Phase
D = Displacement
V = Velocity
A = Acceleration



MAIN MODERATOR PUMPS (2/UNIT)

170614 RD

AUXILIARY MODERATOR PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 2
SPEED = 1,200 RPM
TYPE = HORIZONTAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER =
POWER = PH/HZ/V =
MOTOR WEIGHT =
TYPE =
BEARING TYPE =

PUMP

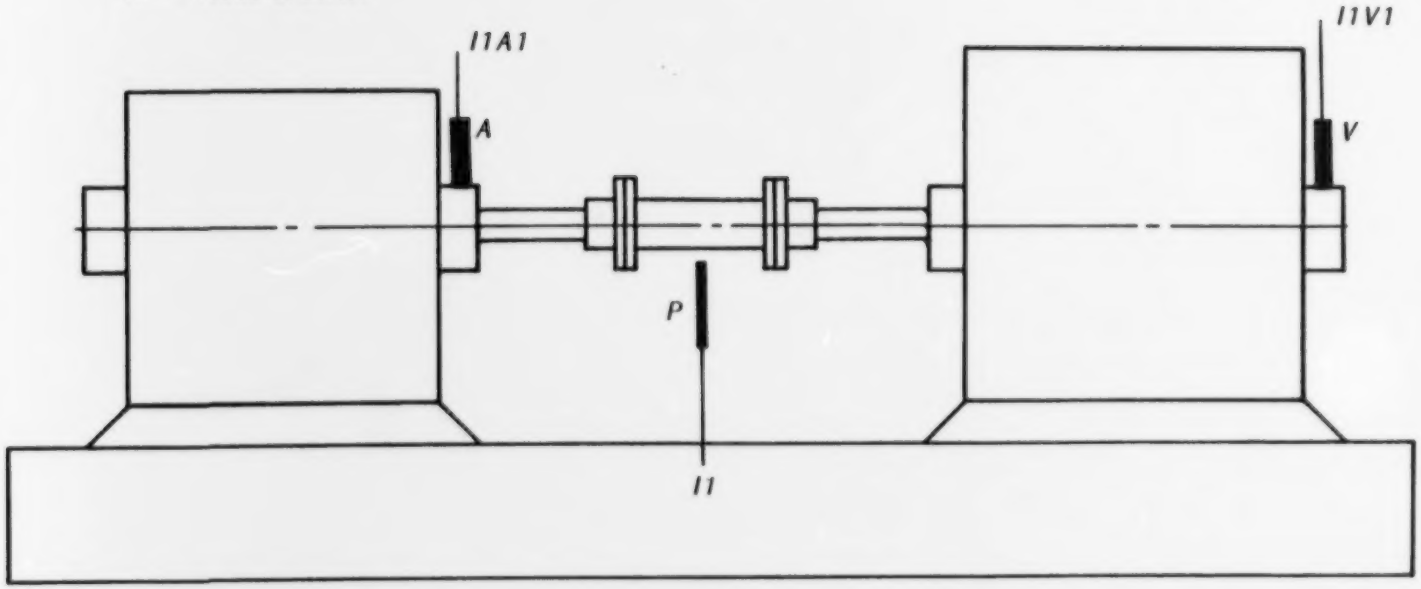
MANUFACTURER =
PUMP SIZE AND TYPE =
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

1 - PHASE
1 - VELOCITY ON MOTOR NON-DRIVE END BEARING
1 - ACCELERATION ON PUMP DRIVE BEARING

Pump 1

P = Phase
V = Velocity
A = Acceleration



AUXILIARY MODERATOR PUMPSETS (2/UNIT)

170615 RD

PRIMARY HEAT TRANSPORT PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 4
SPEED = 1,800 RPM
TYPE = VERTICAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER = SIEMENS A.G.
POWER = PH/HZ/V =
MOTOR WEIGHT =
TYPE =
BEARING TYPE =

PUMP

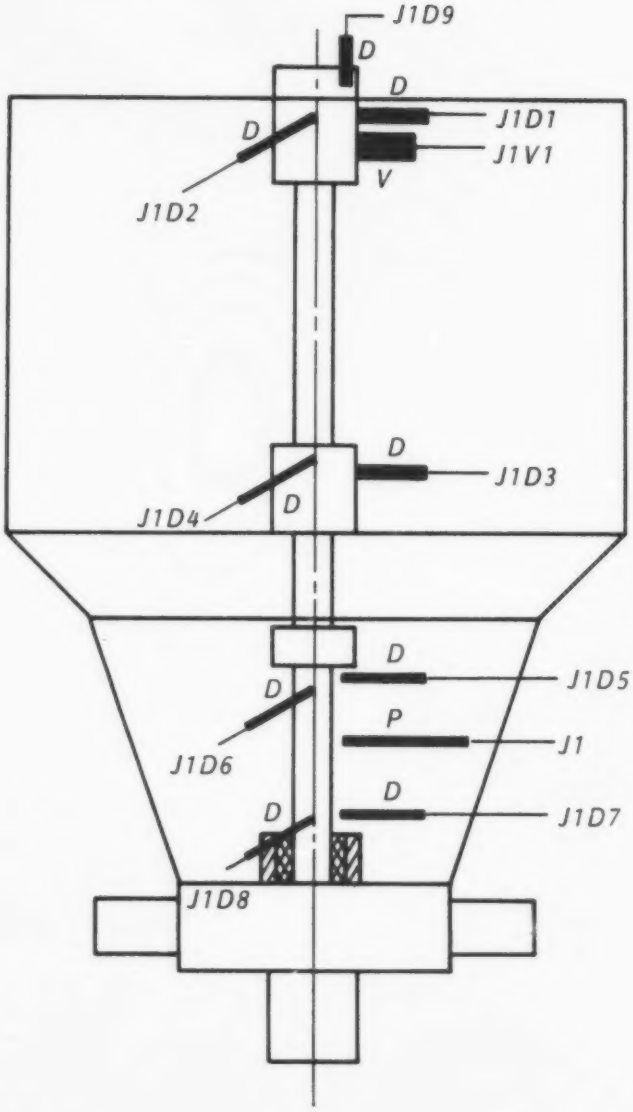
MANUFACTURER = BYRON JACKSON
PUMP SIZE AND TYPE = 20 x 20 x 24
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

1 - PHASE
9 - DISPLACEMENT ON MOTOR TOP/BOTTOM BEARINGS, MOTOR THRUST BEARING, ABOVE PUMP SEAL AND BELOW PUMP COUPLING
1 - VELOCITY ON TOP OF MOTOR<

Pump J

P = Phase
D = Displacement
V = Velocity



PRIMARY HEAT TRANSPORT PUMPS (4/UNIT)

PRIMARY HEAT TRANSPORT FEED PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 2
SPEED = 3,600 RPM
TYPE = HORIZONTAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER = BROWN BROVERI
POWER = 700 KW
PH/HZ/V = 3/60/4000
MOTOR WEIGHT = 3,400 KG
TYPE =
BEARING TYPE = SLEEVE

PUMP

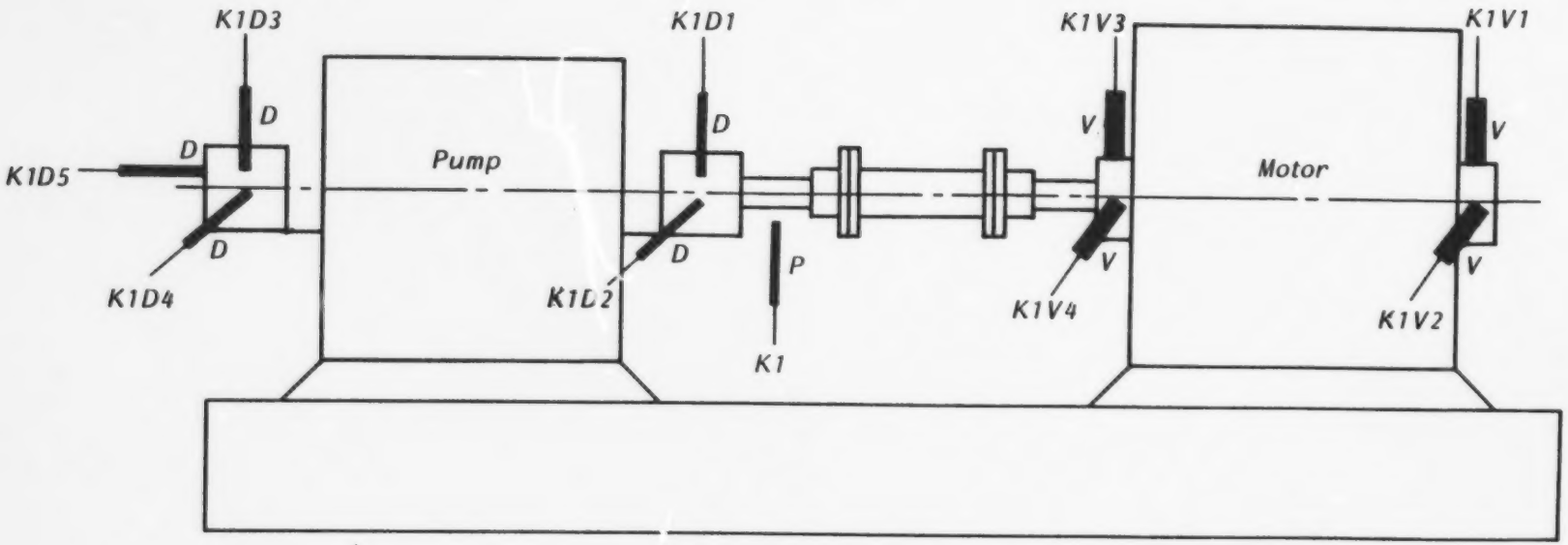
MANUFACTURER =
PUMP SIZE AND TYPE =
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

1 - PHASE
5 - DISPLACEMENT ON PUMP INBOARD/OUTBOARD BEARING AND THRUST BEARING
4 - VELOCITY ON MOTOR INBOARD/OUTBOARD BEARINGS

Pump K

P = Phase
D = Displacement
V = Velocity



PRIMARY HEAT TRANSPORT FEED PUMPS (2/UNIT)

170617 RD

SHUTDOWN COOLING PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 3
SPEED = 1,800 RPM
TYPE = VERTICAL
TOTAL WEIGHT = 9,500 KG

MOTOR

MANUFACTURER = RELIANCE ELECTRIC
POWER = 270 KW PH/HZ/V =
MOTOR WEIGHT = 4,500 KG
TYPE =
BEARING TYPE = ANTIFRICTION TOP - FAG - QJ 332 NP MPA
BTM - SKF NU316

PUMP

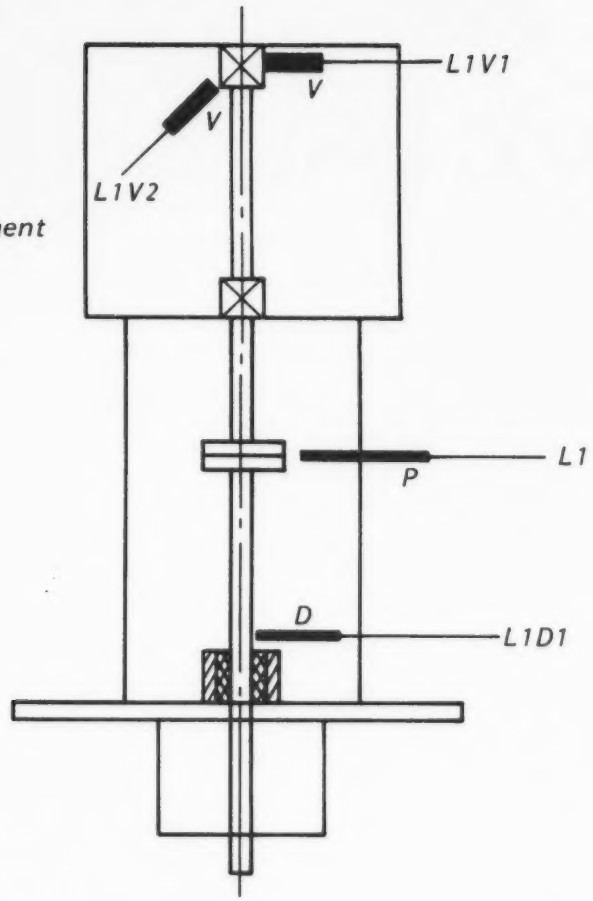
MANUFACTURER = BINGHAM WILLAMETTE
PUMP SIZE AND TYPE = 8 x 10 x 17 RV
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES = 1
PUMP WEIGHT = 5,000 KG
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

1 - PHASE
1 - DISPLACEMENT ABOVE PUMP SEAL
2 - VELOCITY ON MOTOR TOP BEARING

Pump L

P = Phase
D = Displacement
V = Velocity



SHUTDOWN COOLING PUMPS (3/UNIT)

H.P. ECI PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 2 PER STATION
SPEED =
TYPE = HORIZONTAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER =
POWER = PH/HZ/V =
MOTOR WEIGHT =
TYPE =
BEARING TYPE =

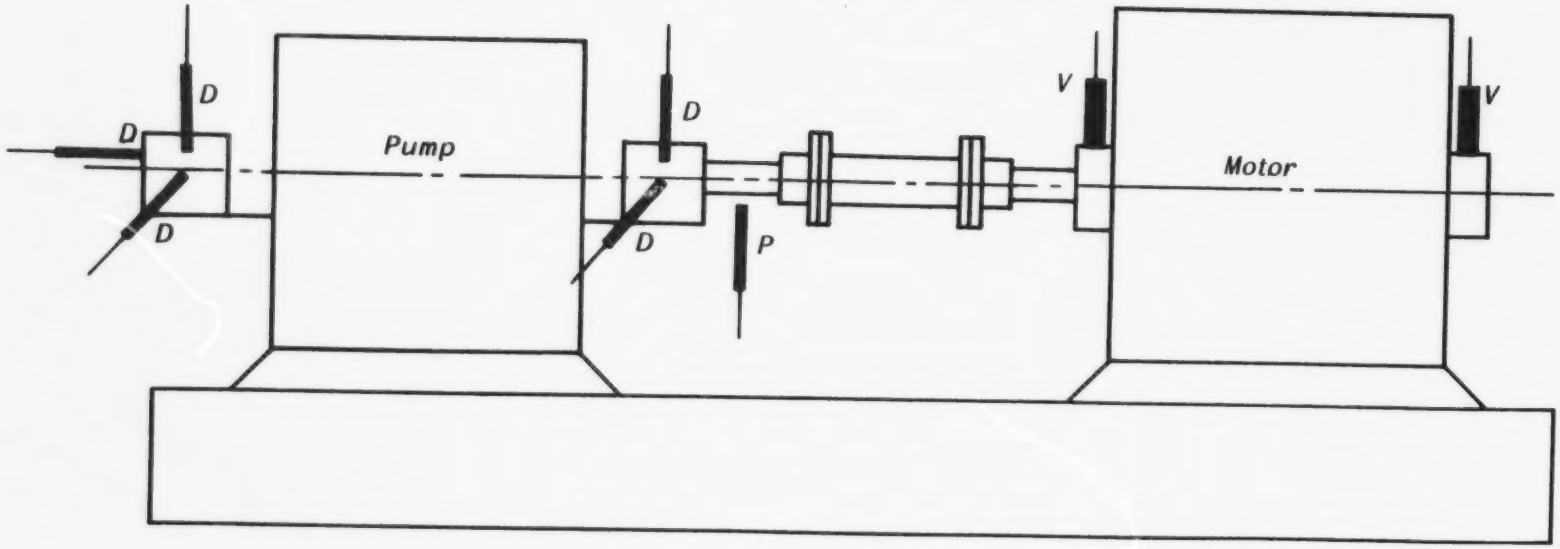
PUMP

MANUFACTURER =
PUMP SIZE AND TYPE =
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

1 - PHASE
5 - DISPLACEMENT ON PUMP INBOARD/OUTBOARD BEARINGS AND ON THRUST BEARING
2 - VELOCITY ON MOTOR INBOARD/OUTBOARD BEARINGS

P = Phase
D = Displacement
V = Velocity



H.P. ECIS PUMPS (2/STATION)

170619 RD

L.P. ECI PUMPSETS

GENERAL INFORMATION

NUMBER OF PUMPSETS PER UNIT = 6 PER STATION
SPEED =
TYPE = VERTICAL
TOTAL WEIGHT =

MOTOR

MANUFACTURER =
POWER = PH/HZ/V =
MOTOR WEIGHT =
TYPE =
BEARING TYPE =

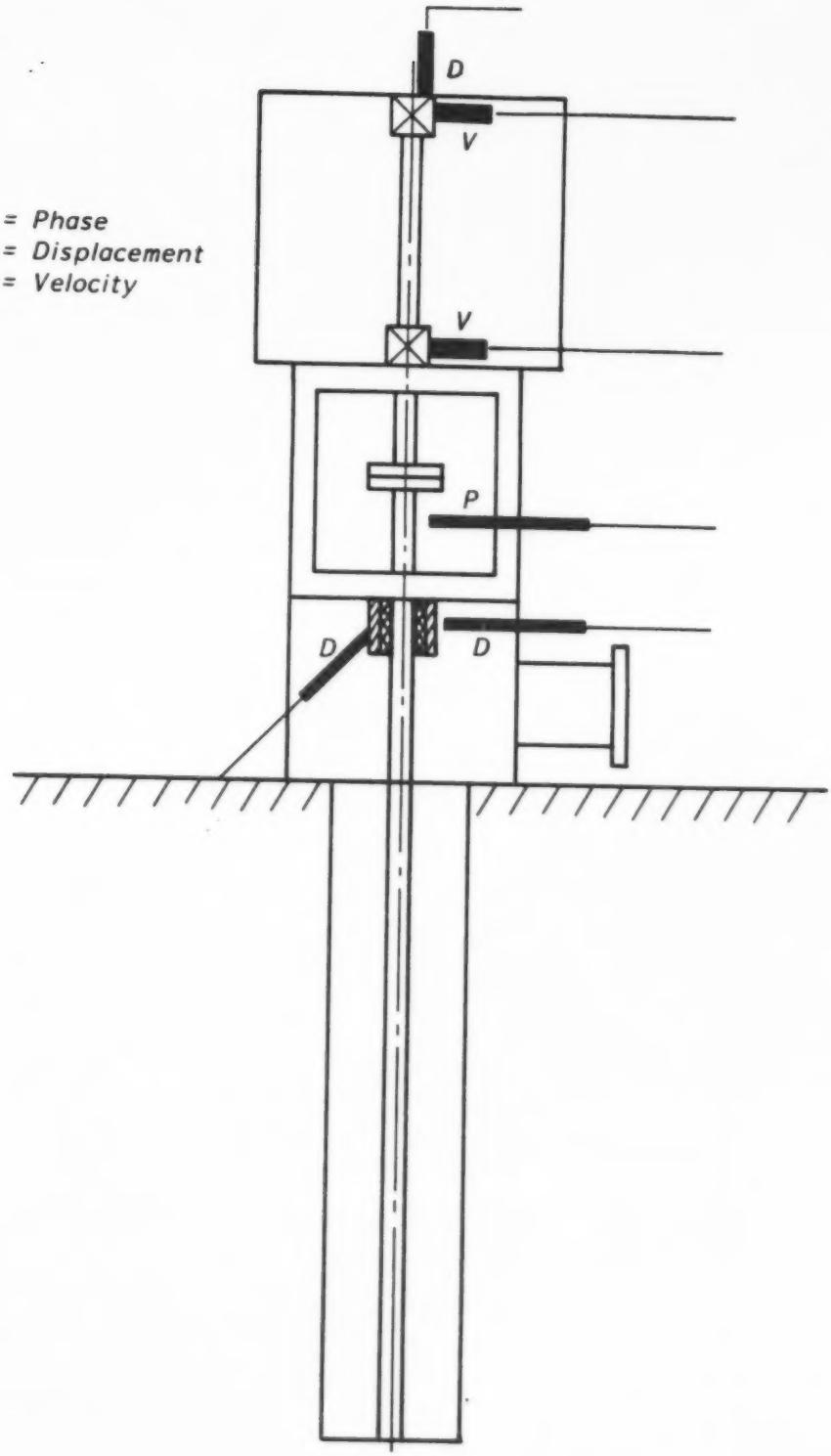
PUMP

MANUFACTURER =
PUMP SIZE AND TYPE =
SEAL SIZE AND TYPE =
BEARING TYPE =
NUMBER OF BLADES =
NUMBER OF STAGES =
PUMP WEIGHT =
CAPACITY =
DESIGN HEAD =
COUPLING =

TRANSDUCERS

1 - PHASE
3 - DISPLACEMENT ON PUMP ABOVE STUFFING BOX AND ON MOTOR THRUST BEARING
2 - VELOCITY ON MOTOR TOP AND BOTTOM BEARINGS

P = Phase
D = Displacement
V = Velocity



LP ECIS PUMPS 6/STATION

170620 RD

APPENDIX B

IMPLEMENTATION OF DIGITAL FILTERS

The method selected for the development of the digital filters allows for the design of a class of butterworth high pass, low pass, bank pass or band stop filters, from specific pass band and stop band edges, desired attenuation and allowable ripple, in an algorithm form.

The most frequently used approximation method for the design of recursive digital filters is the bilinear transformation method. In this method a discrete transfer function $H_D(Z)$ is derived from a corresponding continuous transfer function H(S) as

$$H_D(Z) = H(S)\Big]_{S = 2(Z-1)/T(Z+1)} \qquad (B1)$$

where S is the Laplace variable
Z is the Z variable
T is the sampling period

If H(S) represents a low pass, band pass etc analog filter, then $H_D(Z)$ represents a low pass, band pass etc digital filter.

If the frequency variable in H(S) is denoted as ω and that in $H_D(Z)$ as Ω then $\left|H_D(e^{j\Omega T})\right| = \left|H\ (j\omega)\right|$ provided that $\omega = w2/T \tan (\Omega T/2)$.

For $\Omega << \omega_s/2$, where ω_s is the sampling frequency, $\omega \approx \Omega$ and hence the derived digital filter will have the same amplitude response as the analog filter at approximately the same frequency. As Ω is increased, however, the relation between ω and Ω becomes nonlinear and the frequency scale for the digital filter will be distorted in relation to that of the analog filter. This is known as the warping effect.

If only the amplitude response of the digital filter is of concern the warping effect can for all practical purposes be eliminated by "pre-warping" H(S).

With the prescribed pass band and stop band edges for the digital filter denoted as Ω_i for i = 1, 2, 3, ... pre-warping can be effected by choosing the corresponding pass band and stop band edges in H(S) as

$$\omega_i = \frac{2}{T} \tan\left(\frac{\Omega_i T}{2}\right) \qquad \text{(B2)}$$

for i = 1,2,3, ...

The design formulas used in the development of the butterworth-type digital filter are given in reference/ /.